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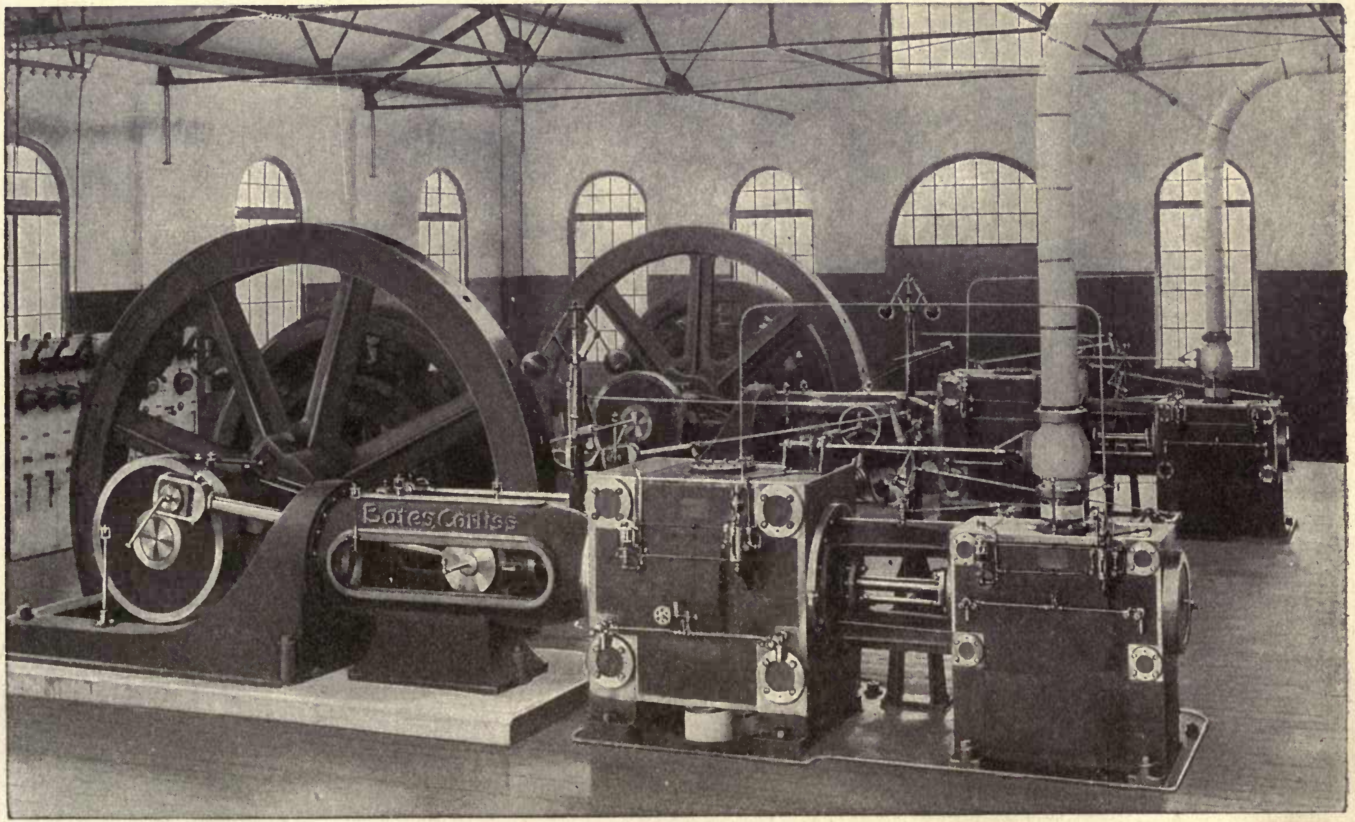

TANDEM COMPOUND BATES CORLISS ENGINES DIRECT-CONNECTED TO 400 K.W. GENERATORS.

Power Station of the Hampton Roads Railway Co., Hampton, Va.

Valve Gears and Indicators

A Manual of

PRACTICAL INSTRUCTION IN VALVE-SETTING, USE OF INDICATORS, AND OTHER DETAILS OF STEAM ENGINE OPERATION ESSENTIAL TO EFFICIENCY AND ECONOMY

PART I—VALVE GEARS

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PART II—STEAM ENGINE INDICATORS

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Foreword

IN recent years, such marvelous advances have been made in the engineering and scientific fields, and so rapid has been the evolution of mechanical and constructive processes and methods, that a distinct need has been created for a series of *practical working guides*, of convenient size and low cost, embodying the accumulated results of experience and the most approved modern practice along a great variety of lines. To fill this acknowledged need, is the special purpose of the series of handbooks to which this volume belongs.

¶ In the preparation of this series, it has been the aim of the publishers to lay special stress on the *practical* side of each subject, as distinguished from mere theoretical or academic discussion. Each volume is written by a well-known expert of acknowledged authority in his special line, and is based on a most careful study of practical needs and up-to-date methods as developed under the conditions of actual practice in the field, the shop, the mill, the power house, the drafting room, the engine room, etc.

¶ These volumes are especially adapted for purposes of self-instruction and home study. The utmost care has been used to bring the treatment of each subject within the range of the com-

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mon understanding, so that the work will appeal not only to the technically trained expert, but also to the beginner and the self-taught practical man who wishes to keep abreast of modern progress. The language is simple and clear; heavy technical terms and the formulæ of the higher mathematics have been avoided, yet without sacrificing any of the requirements of practical instruction; the arrangement of matter is such as to carry the reader along by easy steps to complete mastery of each subject; frequent examples for practice are given, to enable the reader to test his knowledge and make it a permanent possession; and the illustrations are selected with the greatest care to supplement and make clear the references in the text.

¶ The method adopted in the preparation of these volumes is that which the American School of Correspondence has developed and employed so successfully for many years. It is not an experiment, but has stood the severest of all tests—that of practical use—which has demonstrated it to be the best method yet devised for the education of the busy working man.

¶ For purposes of ready reference and timely information when needed, it is believed that this series of handbooks will be found to meet every requirement.

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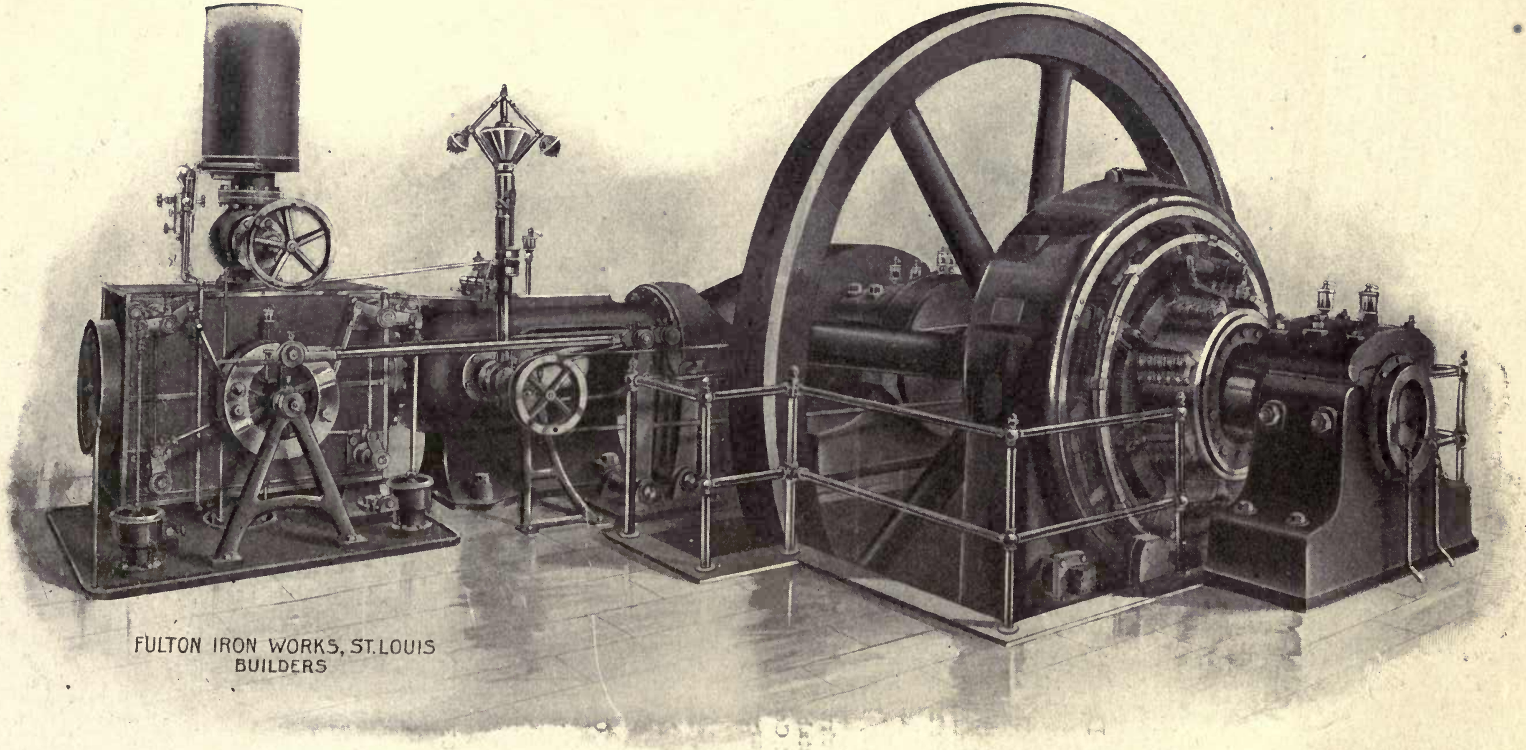

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VALVE GEARS

Steam enters the cylinder of the engine through ports which must, in some manner, be opened and closed alternately, in order to admit and exhaust the steam at the proper time. To accomplish this purpose a valve is moved back and forth across the port openings. A complete understanding of the valve and valve gear is essential to the engineer as well as to the designer, for even though a valve be properly designed, its economy may be seriously impaired by improper setting. The design and adjustment of these valves plays a very important part in the efficient action of the steam engine.

The term "valve gear" includes the valve or valves that admit steam to and exhaust it from the cylinder of the engine, together with the mechanism from which the valves derive motion. There may be a single valve to regulate admission and exhaust, or there may be a double set of valves; one set to admit the steam at each end and another to release it. The valve may have a plain reciprocating motion, moved by a rod, or it may be opened by some device that lets go at the proper time, allowing the valve to drop shut under the influence of counter weights, springs or vacuum dashpots. To the first class belong the plain slide valve and its modification of piston valve, gridiron valve, etc.; to the second belong such valves as the Corliss, Brown, and others.

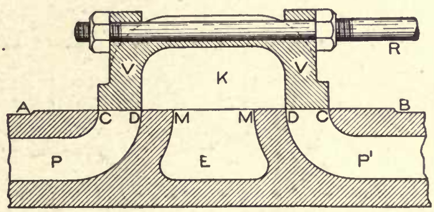


Fig. 1.

The simplest type of valve is the plain slide or D valve as shown in Fig. 1.

In this figure V is the valve, R the valve rod, K the exhaust cavity, P and P′ the steam ports, E the exhaust port, AB the valve seat, and DM the bridges of the valve seat. The valve seat must be planed perfectly smooth, so that pressure on the valve will

make a steam tight fit, and cause as little friction as possible when the valve slides. Furthermore, the length of the seat AB must be a little less than the distance from the extreme right-hand position of the right-hand edge of the valve to the extreme left-hand position of the left-hand edge of the valve. This allows the valve at each stroke slightly to over travel the seat, thus keeping it always worn perfectly flat and smooth. If the valve seat were not raised slightly above the rest of the casting, or if it were too short, the constant motion of the valve would soon wear a hollow path in the valve seat, and it would cease to be steam tight.

Eccentric. The valve usually receives its motion from an eccentric which is simply a disc, keyed to the shaft in such a

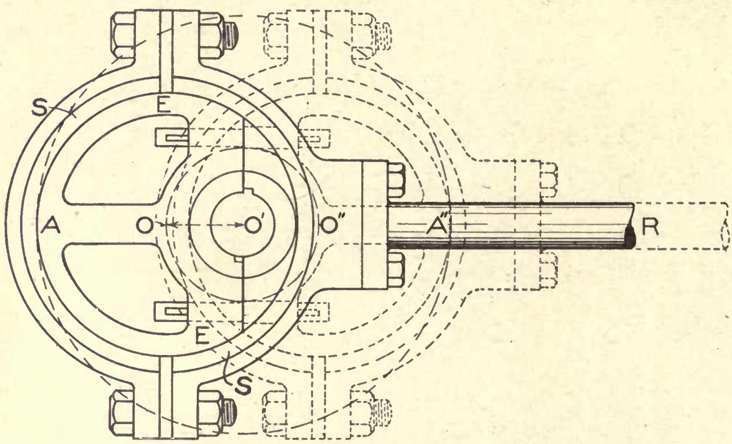


Fig. 2.

manner that the center of the disc and the center of the shaft do not coincide. It is evident that as the shaft revolves, the center of this eccentric disc moves in a circle about the shaft as a center, just as if it were at the end of a crank. The action of the eccentric is equivalent to the action of a crank the length of which is equal to the eccentricity of the eccentric (the distance between the center of the eccentric and that of the shaft).

Fig. 2 represents the essentials of an ordinary eccentric. O is the center of the shaft, O the center of the eccentric disc E, and S is a collar encircling the eccentric and attached to the valve rod R.

As the eccentric turns in the strap, the point O moves in the dotted circle around O′, and the point A also moves in a circle. When half a revolution is accomplished the point O will be at O″, the point A will be at A″, and the eccentric strap and valve rod will be in the position indicated by the dotted lines.

Since the revolving shaft transmits motion to the valve through the eccentric, it will be necessary to study the relative motions of the crank and eccentric in order to get a clear idea of the steam distribution.

The distance of the center of the eccentric from the center of the shaft (OO′ in Fig. 2) is known as the eccentricity, or throw, of the eccentric. The travel of the valve is twice the eccentricity.

Valve without Lap. Fig. 3 shows a section through the steam and exhaust ports of an engine, together with a plain slide

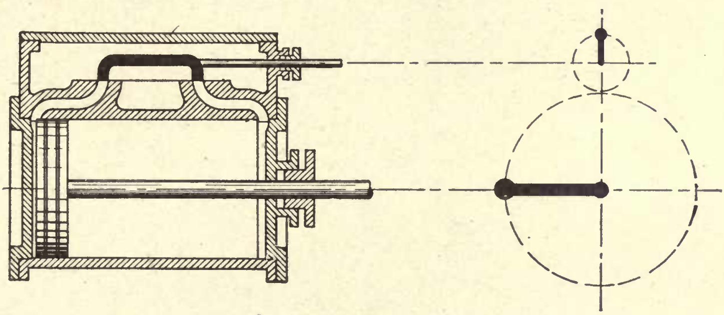


Fig. 3.

valve placed in mid-position, and so constructed that in this position it just covers the steam ports and no more. A valve is in mid-position when the center line of the valve coincides with the center line of the exhaust port.

Fig. 1 shows the same valve drawn to a larger scale.

Suppose the valve is moved a slight distance to the right; the port P (see Fig. 1) is then uncovered and opened to the live steam which enters the cylinder and causes the piston to move. Since the two faces of the valve are just sufficient to cover the steam ports, it is evident that as the port P opens to live steam, the port P′ opens to the exhaust. The ports are closed only when the valve is in mid-position. This allows admission and exhaust

to continue during the whole stroke. With such a valve there is no expansion or compression; the indicator card would be a rectangle, and the M. E. P. would be equal to the initial steam pressure, assuming no frictional losses in the steam pipe or condensation in the cylinder.

For a theoretical discussion of valve motion, it is assumed that the eccentric rod moves back and forth in a line parallel to the center line of the engine. This is not the case in practice, for the eccentric rod always makes a small angle with the center line, just as the connecting rod does, but the eccentricity is so small in comparison with the length of the eccentric rod that the angularity of the eccentric rod is very much smaller than the angularity of the connecting rod, and its influence may be neglected without appreciable error.

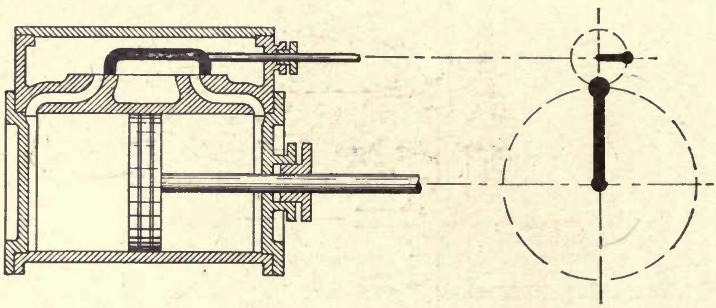
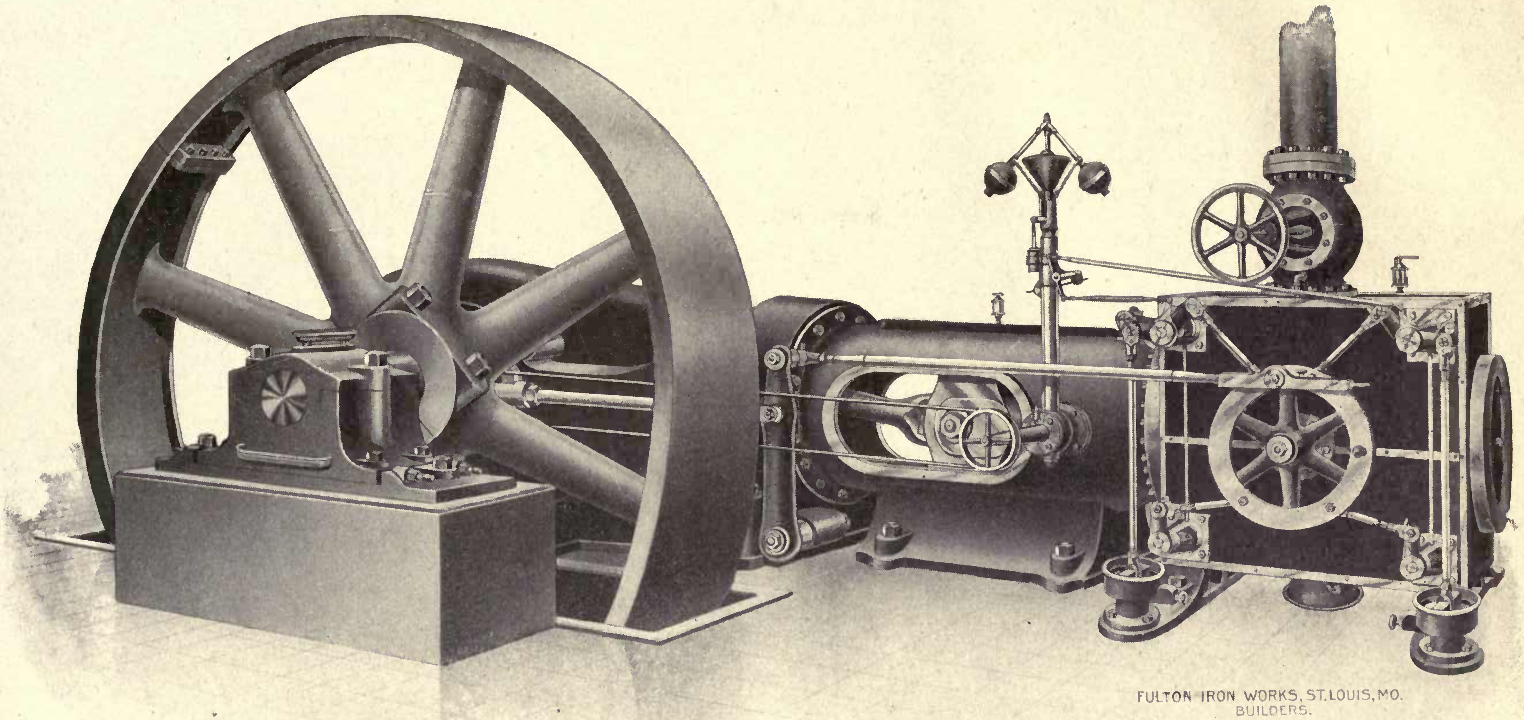


Fig. 4.

When the valve shown in Fig. 3 is in mid-position, the crank is on dead center, the eccentric is set at right angles to it, and the piston is just ready to begin the stroke.

Fig. 4 shows the relative positions of crank, piston, eccentric and valve when the crank has made a quarter turn or the piston has moved to half stroke. The eccentric is now in its extreme position to the right, the valve has its maximum displacement and both the steam and exhaust ports are wide open. The valve will not close again until the piston has reached the end of its stroke.

This type of valve is used only on small and unimportant engines, and since it allows no expansion of the steam, is very uneconomical. Furthermore, it will be seen that this valve opens just after the stroke begins, which is impractical, for it means that the piston has begun its stroke before the full steam pressure



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reaches it, which will cause an inclined admission line on the indicator diagram.

Valve with Lap. If the face of the valve is made longer than shown in Fig. 1, so that in mid-position it overlaps the steam ports, we shall have a valve such as shown in Fig. 5. The amount that the valve overlaps the steam ports is called the lap of the valve. In Fig. 5, DI is the *inside* lap, and OC the *outside* lap. It will at once be seen that both the admission and exhaust ports may remain closed during a part of the stroke, thus making expansion

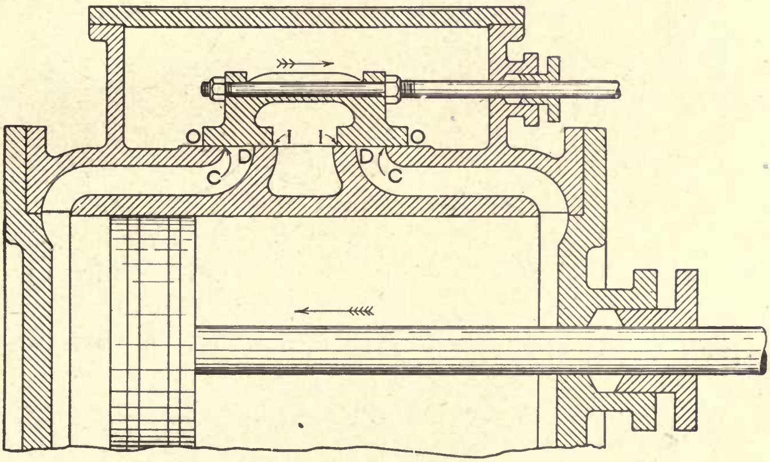

Fig. 5.

and compression possible. It is also evident that steam cannot be admitted until the valve uncovers the port by moving a distance from mid-position equal to OC. Admission continues until the valve returns to such a position that the outer edge of the valve again closes the port. Release will begin when the inner edge of the inside lap begins to uncover the port.

Fig. 6 represents a valve with lap, at the point of admission. Since the valve must move a distance equal to the outside lap before admission can take place, it is evident that the eccentric can no longer be at right angles to the crank at the beginning of the stroke, but must be ahead of the right-angle point by an amount equal to AOC. The angle AOC is known as the angular advance.

The maximum displacement of the valve is attained when the eccentric is horizontal as shown in Fig. 7. In this position

both the steam and exhaust ports are wide open, and any further motion of the piston will cause the valve to move toward its mid-position.

Admission continues until the valve returns to the position

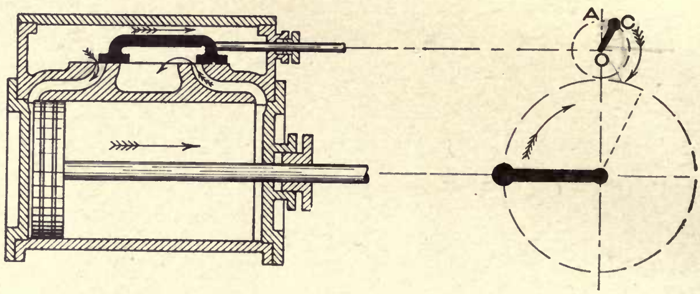

Fig. 6.

shown in Fig. 8. Here the outside lap just closes the left-hand steam port, cut-off takes place, and the steam already in the cylinder begins to expand. As the valve continues to move toward the left, the left-hand inside lap begins to uncover the left-hand port and releases the steam at the position shown in Fig. 10.

The dotted lines of Fig. 7 show the valve in its extreme

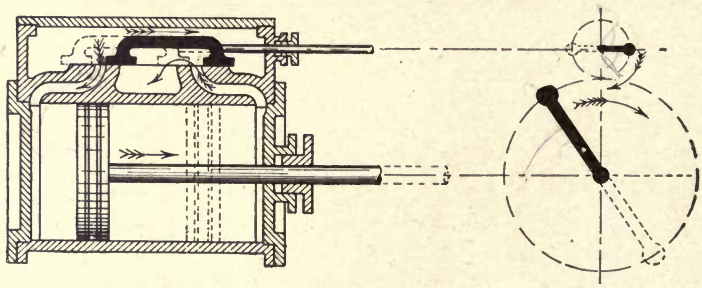

Fig. 7.

position to the left. Any further movement of the piston will cause it to return toward mid-position.

The dotted position of crank and eccentric in Fig. 10 shows the valve returned to the point of compression, which continues until the conditions of Fig. 6 are again reached and the opening valve allows steam again to enter the cylinder.

This process has been traced step by step for one end only; let us now consider what is happening at the other end.

Admission is the point at which the valve opens to admit steam to the cylinder. **Cut-off** is the point at which the valve closes to cut off the admission of steam. **Release** is the point at which the exhaust is opened; and **Compression** is the point at which the exhaust is closed.

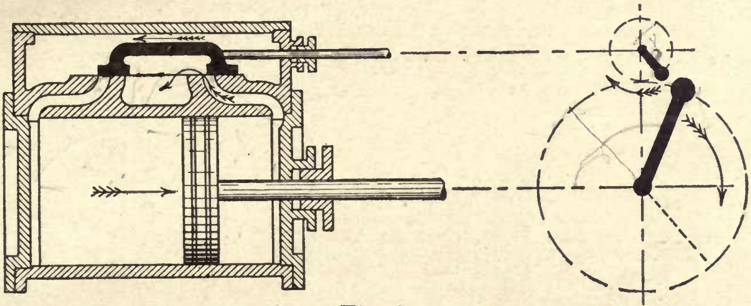

Fig. 8.

While the crank is moving from the position shown in Fig. 6 to that of Fig. 8, steam is being admitted to the head end and being exhausted from the crank end. The inside lap being less than the outside lap, causes the exhaust to continue longer than the admission.

Fig. 9 shows the relative positions of crank, eccentric and valve when the exhaust closes on the crank end and compression

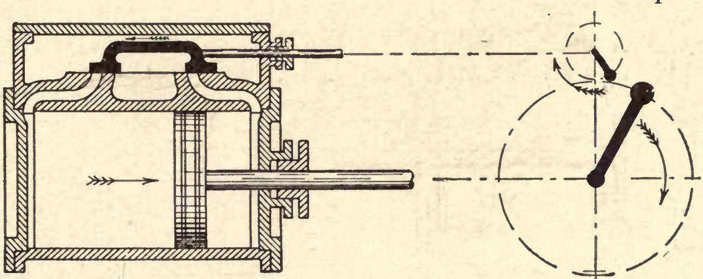

Fig. 9.

begins. Between these two positions the steam is expanding in the head end and exhausting from the crank end.

Between the positions of Fig. 9 and Fig. 10 both ports are entirely closed, expansion is taking place in the head end and compression in the crank end. Fig. 10 is head-end release. Fig. 11 shows admission at crank end of cylinder and marks the end of crank-end compression.

By referring to Figs. 6–11, the effect of any change of lap may at once be observed. If the outside lap is *increased*, the valve must move farther from mid-position before admission will occur, and on the return, after the maximum displacement is reached, the outside lap, being wider, will close the port sooner, and the cut-off shown in Fig. 8 will take place before the crank

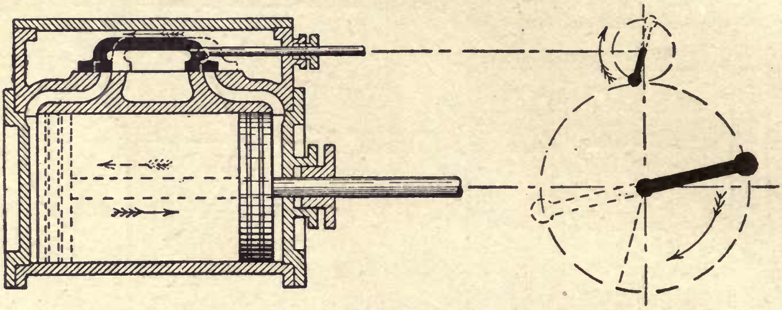

Fig. 10.

reaches the angle there shown. A decrease of outside lap will make cut-off later and admission earlier.

If the inside lap is increased, the valve must move farther before release occurs and the crank angle would be greater than shown in Fig. 10. On the return of the valve to the dotted position shown in Fig. 10, the port will close earlier and make an earlier compression; the crank angle will be less than is there shown. Decreasing inside lap will cause earlier release and later compression.

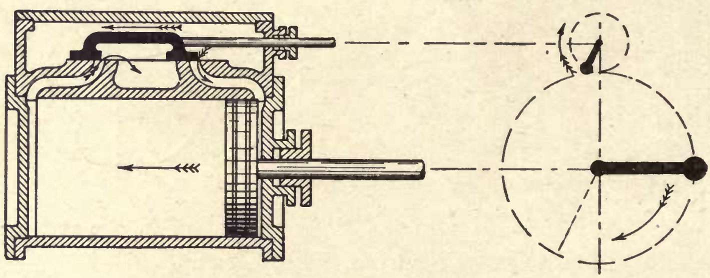

Fig. 11.

Thus we see that it is the outside lap that influences admission and cut-off, and the inside lap that controls release and compression. For this reason the outside lap is often called the *steam lap*, and the inside lap the *exhaust lap*.

Lead. If a valve having lap is in mid-position, the port is closed and the engine cannot start because no steam can enter the cylinder. That the steam may be ready to enter the cylinder at the beginning of the stroke it is necessary that the eccentric be set more than 90° ahead of the crank and the eccentric radius will take an angle as shown in Fig. 6, called the *angular advance.* In order that the ports and clearance may be properly filled with steam at the beginning of the stroke, it is necessary that the valve be displaced from its mid-position an amount slightly greater than the outside lap. With the piston at the end of the stroke the valve will have a position as shown in Fig. 12. The port will be open the distance AB. This causes the eccentric to be moved forward a slight amount in excess of the angular advance. This excess is called the *angle of lead.*

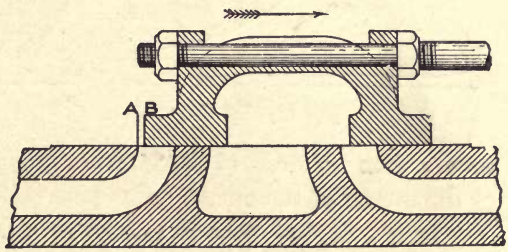

Fig. 12.

In Fig. 13, O′R′ represents the crank at the beginning of the stroke, LOA the angular advance, and AOA′ the angle of lead. The eccentric, to give lead, must be set at the angle ROA′ ahead of the crank or 90° plus angular advance plus angle of lead. In large, quick-running engines, a liberal lead is essential, so that the ports and clearance may be well filled with steam before the stroke begins. If there is no lead, a portion of the steam will be used in filling these places and full pressure steam will not reach the piston until it is well advanced on the stroke. This will give a sloping admission line as shown

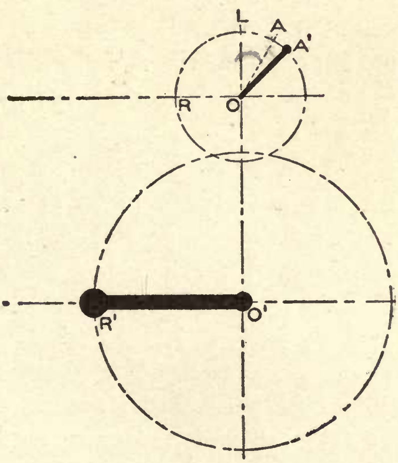

Fig. 13.

in Fig. 14. Too much lead, on the other hand, will cause too early an admission as shown in Fig 15.

If the angular advance is increased, the eccentric will be moved farther ahead of the crank, and consequently will begin its motion sooner. It will necessarily arrive at each of the events

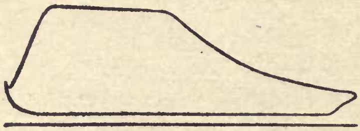

Fig 14.

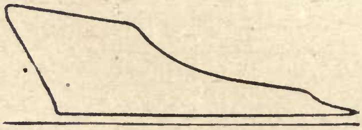

Fig. 15.

sooner than before. If then, the angular advance is increased, all of the events of the stroke will occur earlier.

Inequality of Steam Distribution. In the valve diagrams thus far considered, the events of the stroke have been discussed for each end separately, without reference to the relation of similar events on the other side of the piston. If the connecting rod were of infinite length, so that it would always remain parallel to the center line of the engine, the distribution would be the same for both ends of the cylinder. In practice, the connecting rod is from 4 to 8 times the length of the crank, which causes the connecting rod always to be at an angle to the center line of the engine, and for a given crank angle makes the piston displacement greater at the head end than at the crank end.

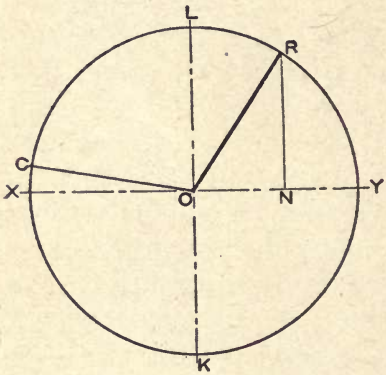

Fig. 16.

To find the displacement of the valve, let us consider Fig. 16. The circle represents the path of the eccentric center during a complete revolution of the engine. OC represents the crank, and OR the corresponding position of the eccentric. The diameter XY represents the extent of the valve travel. Since the eccentric rod is so long in comparison to the eccentricity, we make no appreciable error by assuming it always to be parallel to the center

line of the engine. When the eccentric is at OL, the valve is in mid-position. At OR the valve has moved from mid-position an amount ON, found by dropping a perpendicular from R to the center line XY. If the angularity of the connecting rod could be neglected, the *piston* displacement could be found in the same manner.

To find the displacement of the piston, a diagram as shown in Fig. 17 must be drawn. In this figure AB represents the cylinder, P the piston, H the crosshead, HR the connecting rod, and OR the crank. Suppose now the engine should stop in this position and then be clamped. The piston displacement would be represented by AP. If the crank pin at R should now be loosened so as to allow the connecting rod to fall to a horizontal position, the point R would describe the arc of a circle RN, and XN would represent the piston displacement and would be equal to AP

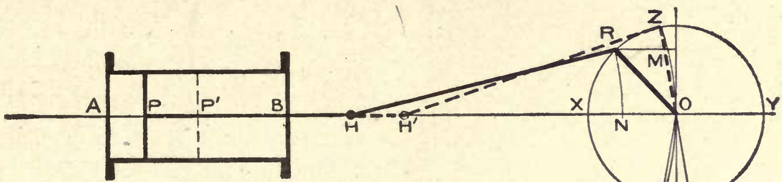

Fig. 17

Suppose now that in this disconnected way the piston, crosshead and connecting rod were moved forward until the end of the rod came to O. P would then be at P′ and the piston would be in the middle of its stroke. Now swing the end of the rod up to its proper position on the crank-pin circle, the piston remaining stationary. It would describe an arc OZ. The crank pin would be at Z, less than a quarter revolution from X, while the piston would be in the middle of its stroke.

Suppose this engine were running with cut-off at half stroke on the head end and that XOZ represented the corresponding crank angle. On the return stroke the valve would cut off at the same crank angle YOT = XOZ, and OT would represent the crank cut-off on the return or crank-end stroke. The piston, as we have just seen, will not be at half stroke except when the crank is at OZ or OS. Consequently OT is less than half stroke and cut-off takes place earlier at the crank end than at the head end. When the crank is at OZ the eccentric will be at OA (Fig 17*a*), and the valve displace-

ment will be OB. When the crank is at OT the eccentric will be at OA′, and the valve displacement will be OB′, which is equal to OB, the displacement of the valve at cut-off in the head end. The piston displacement will be OX in the head end and WY in the crank end when cut-off occurs. If the connecting rod always remained parallel to the center line, the cut-off would be the same at both ends.

Compensation of Cut-off. It has already been pointed out that lengthening the outside lap makes the cut-off earlier, and shortening the lap makes it later. The cut-off in the case just cited may then be equalized by altering the outside laps. If we increase the outside lap on the head end, or decrease the crank-end lap, the inequality will be less. By changing either or both of the laps the proper amount, the cut-off may be exactly equalized.

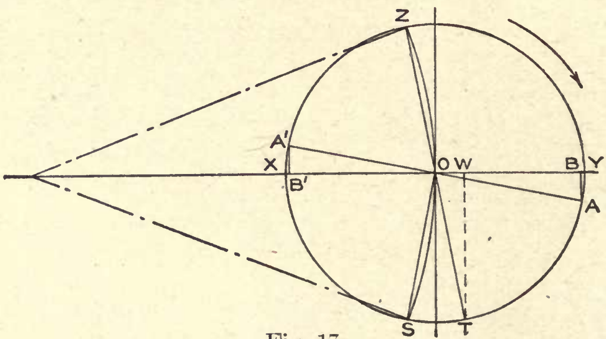

Fig. 17*a*.

But altering the outside lap changes the lead as has already been explained. If the lap is increased on the head end, the lead will be less than on the crank end. If the lead becomes too small on the head end, the angular advance may be increased but the inequality of lead will still remain, for this increase of angular advance will increase the lead at the crank end as well as at the head end, and by hastening all the events of the stroke may give a bad steam distribution if care is not taken.

Unequal lead is of less consequence on a low-speed than on a high-speed engine. On low-speed engines the cut-off may be equalized at the expense of lead with beneficial results, but on high-speed engines it will not do to give too little lead at one end. A high-speed engine requires more lead than a low-speed, for there is relatively less time in each stroke for the clearance to fill with steam.

If both inside laps are equal, compression will not occur equally at both ends. To equalize it, the inside laps may be changed in the same manner as the outside laps are changed to equalize the cut-off. By altering these inside laps to equalize compression, it may happen that the lap is reduced enough to leave the exhaust port open when the valve is in mid-position.

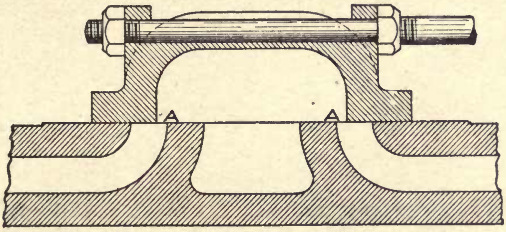

Fig. 18.

This opening of the valve is called an inside clearance, or *negative lap*. In Fig. 18, A is the inside clearance.

Rocker. Sometimes it happens that the valve stem and eccentric rod cannot be so placed that they will be in the same straight line; or it may be that the travel of the valve must be so great as to require an excessively large eccentric. In such cases a rocker may be used.

Fig. 19 shows a valve that is not in line with the eccentric. This occurs in horizontal engines when the valve is set on top of

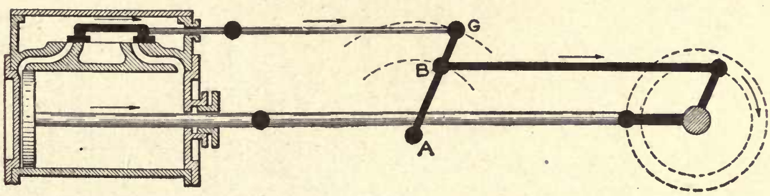

Fig. 19.

the cylinder instead of on one side. By means of the rocker AG the valve may receive its proper motion.

In case it is more convenient to place the pivot of the rocker arm between the connections to the valve stem and those of the eccentric rod, such an arrangement as shown in Fig. 20 may be used. Here it will be noticed that the valve stem and eccentric rod are moving in opposite directions, and to give the valve the

same motion as in Fig. 19, the eccentric must be moved 180° ahead of the position there shown.

If AB is less than AG, the valve travel will be greater than twice the eccentricity, in proportion as AG is greater than AB. In all cases the valve travel is to twice the eccentricity as AG is to AB. Thus, if the valve travel is $4\frac{1}{2}$ inches, AB, 15 inches, and AG, 18 inches, then $\frac{15}{18} \times 4\frac{1}{2} = 3\frac{3}{4}$ inches, will equal twice the eccentricity.

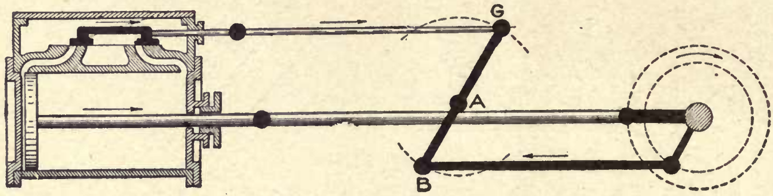

Fig. 20.

A valve gear may be so laid out as to make both the cut-off and the lead equal for both ends of the cylinder. This may be done by a proper porportion between the rocker arms, and a careful location of the pivot of the rocker. The eccentric must then be set accordingly. In this manner the Straight Line engine equalizes the cut-off and lead. A discussion of this method will be considered later.

VALVE DIAGRAMS.

Zeuner's Diagram. In order to study the movements of valves, the effect of lap, lead, eccentricity, etc., diagrams of various sorts have been devised. By the use of diagrams we may acquire a knowledge of valve motion without the complex mathematical expressions that such a discussion would entail. The most useful of these various diagrams is that devised by Zeuner, and to avoid complexity we shall confine ourselves to a discussion of this diagram alone. The eccentric rod is assumed to be of infinite length, and the positions of the crank are shown on the diagrams. The displacement of the piston can easily be found if the ratio of crank to connecting rod is known.

In Fig. 21 let OY be the eccentricity, then XOY will represent the valve travel, and the center of the eccentric will move

in the circle XWY. Let OR represent the position of the crank and Or the corresponding position of the eccentric, which is 90° + angle of advance θ ahead of the crank. Draw OW perpendicular to XY and lay off from it the angle WOM = angle of advance θ towards the crank. With OM as a diameter, construct a circle. OM is equal to the eccentricity, and the circle MPO is known as the *valve circle.* If OR, the center line of

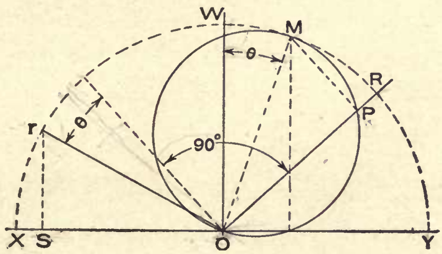

Fig. 21.

the crank, cuts this valve circle at P, then OP is equal to the displacement of the valve from mid-position.

To prove this, draw rS perpendicular to XY. Since Or is the position of the eccentric, OS will represent the valve displacement from mid-position. Draw MP. Then by geometry OPM is a right angle because it is inscribed in a semicircle. OSr is also a right angle ; the two right-angled triangles OSr and OMP are equal because they are similar and have two corresponding sides equal. Or = OM, being radii of the same circle. But we have seen that OS is equal to the valve displacement, therefore OP is also equal to the valve displacement, for it is equal to OS.

Now that the truth of our proposition has been proved, let us see how we may study the valve motion from such a diagram. See Fig. 22. As before, let XY represent the valve travel, then the circle XEYF will represent the path of the center of the eccentric. Let θ be the angular advance and lay off EO toward the crank, making an angle θ with the vertical. Produce EO to F, and on OE and OF as diameters draw the valve circles as shown. Let the outside lap be an amount equal to OV, then with O as a center and OV as a radius draw an arc intersecting the upper valve circle at V and K. Lay off OP equal to the inside lap and

with O as center and OP as a radius draw an arc intersecting the valve circle at P and Q. Draw the crank line AO passing through V. Then, when the crank is in this position, the displacement of the valve is equal to OV (the outside lap) and the steam is ready to enter the cylinder. This is the position of the crank at admission, and the crank angle XOA is called the lead angle. The valve has lead, therefore the admission takes place before the end of the stroke. When the crank reaches the position OE, the displacement of the valve is equal to OE, the eccentricity, and is

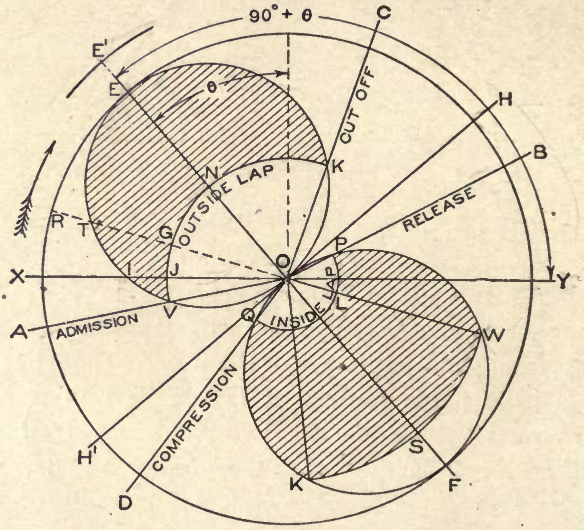

Fig. 22.

the maximum displacement. Further motion of the piston causes the valve to move toward mid-position until, at the crank position OC, the displacement OK is again equal to the outside lap and the valve has reached the point of cut-off. When the position OH is reached, the crank line is tangent to both valve circles and there is no displacement of the valve. At this point the valve is in mid-position.

Further crank movement draws the inside lap toward the edge of the exhaust port until, at the crank position OB, the displacement is equal to OP (the inside lap) and release begins. At OF the maximum valve displacement is again reached and the valve moves in the opposite direction until at OD its displacement

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from mid-position, is again equal to OQ = OP = the inside lap, and compression takes place. At OH′ the valve is again in mid-position. At OX the displacement of the valve is OI, but since the valve has to move the distance OJ before the port begins to open IJ must represent the port opening when the crank is on dead center and by definition we know that lead is the amount of port opening at this position. Therefore IJ represents the lead.

At the position R, the port is open an amount equal to TG, at E the opening is a maximum equal to EN. At C the opening is nothing. If LW represents the total width of the steam port, the exhaust port will be open wide when the displacement of the valve is equal to OW and it will remain wide open while the crank swings from OW to OK.

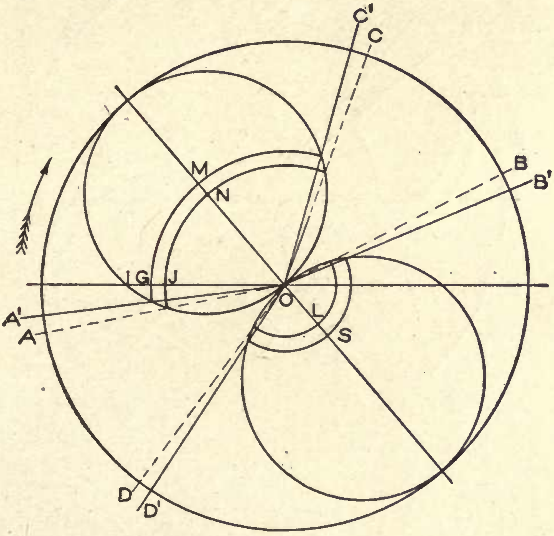

Fig. 23.

If the width of steam port in addition to the outside lap were laid off on the other valve circle it would fall at E′. For the admission port to be wide open, the displacement of the valve would have to be equal to OE′, which is more than the maximum displacement. This shows that in this case the steam port is never fully open and that the left-hand edge of the valve overlaps the right-hand edge of the port by an amount equal to EE′ when the valve has reached its maximum displacement.

Fig. 22, with its two valve circles, shows the diagram for the head end of the cylinder only. The crank-end diagram would be similar except that the laps might not be equal to those of the head end.

We are now in a position to consider more in detail the effect of changing in any way either the valve or the setting. Let us consider Fig. 23, which is in every way like Fig. 22 except that all unnecessary letters and lines are omitted to avoid confusion. If the outside lap is increased an amount equal to NM, the admission will take place later, at crank position OA′; the lead will

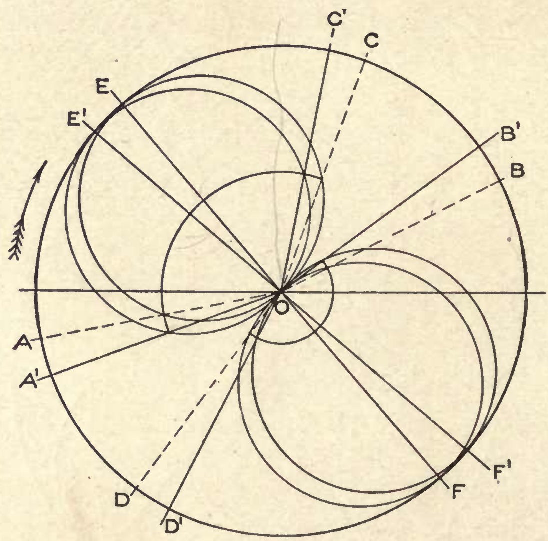

Fig. 24.

be reduced to IG and cut-off will take place earlier at OC′. If the outside lap is *reduced* a like amount the contrary effects will be observed. If the inside lap is increased an amount equal to LS, the release will take place later at the crank position OB′ and compression will take place earlier at OD′. The contrary effect will be observed by decreasing the inside lap.

If the angular advance is increased (see Fig. 24) all the events will occur earlier. This is evident from the figure; the crank revolves in the direction indicated by the arrow and OA′ (new position of admission) is *ahead* of OA, the old position.

If the eccentricity is increased, Fig. 25, the valve travel will increase and admission will take place earlier at OA′; the lead will be increased an amount equal to II′, and cut-off will take place later at OC′. Release will be earlier at OB′ and com-

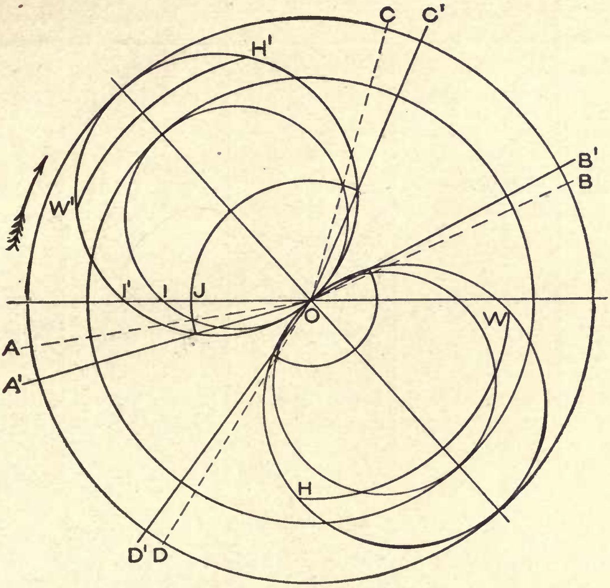

Fig. 25.

pression will be later at OD′. The upper valve circle will now cut the arc drawn from O as a center, with a radius equal to the outside lap plus the width of steam port, in the points W′ and H′, and the admission port will be open wide while the crank is moving from OW′ to OH′. Similarly, the lower valve circle cuts the arc drawn from O as a center, with a radius equal to the inside lap plus the width of steam port, in the points W and H. The steam port is then wide open to exhaust while the crank is moving from W to H. From the above it will be seen that the periods are all changed by changing the travel; thus, admission and exhaust begin sooner and last longer, while expansion and compression begin later and cease sooner. With change in the angular advance, however (see Fig. 24), the periods are neither increased nor decreased.

For convenience, these results are collected in the following table which shows the effect of changing the laps, travel, and angular advance:

	Increasing Outside Lap.	Increasing Inside Lap.	Increasing Travel.	Increasing Angular Advance.
Admission.	Is later. Ceases sooner.	Not changed.	Begins earlier. Continues longer.	Begins earlier. Same period.
Expansion.	Is earlier. Continues longer.	Beginning unchanged. Continues longer.	Begins later. Ceases sooner.	Begins earlier. Same period.
Exhaust.	Unchanged.	Occurs later. Ceases sooner.	Begins earlier. Ceases later.	Begins earlier. Same period.
Compression	Begins at same point.	Begins sooner. Continues longer.	Begins later. Ceases sooner.	Begins earlier. Same period.

PROBLEMS.

All the problems on valve gears involve the relations between certain variables which are:

The valve travel.
Angle of lead.
Outside lap.
Inside lap.
Points of stroke at which admission cut-off, release and compression take place.

In designing a Slide Valve, a few of these variables depend upon the conditions under which the engine is to run. For instance, the valve travel is limited, cut-off must be at a certain point and the engine must have a certain lead. Then, with the aid of a Zeuner's diagram, the remaining proportions of the valve may be determined.

Let us consider a few examples:

Given the valve travel = 3 inches.
Inside lap = ¾ inch.
Angular advance = 35°
Angle at cut-off = 115°

To determine the laps, the lead and the crank angles at admission, compression and release.

In Fig. 26, let XY represent the valve travel = 3 inches. Draw OM perpendicular to XY, and on XY as a diameter draw the circle XMYF representing the path of the center of the eccentric as it revolves about the shaft. Lay off the angle MOE = the angular advance = 35° so that the angle XOE is equal to 90°

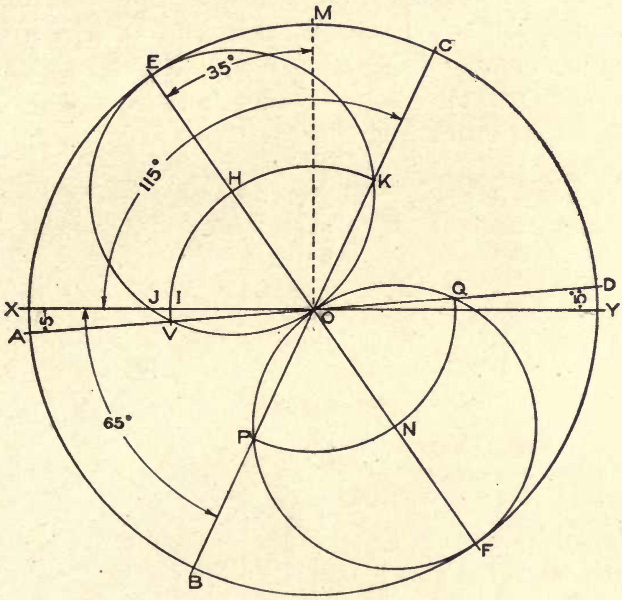

Fig. 26.

minus the angular advance. Produce EO to F. Then on OE and OF as diameters draw the valve circles. The eccentricity OE or OF, if no rocker is used, will be half the valve travel. Lay off the crank angle XOC = angle of crank at cut-off = 115°, and OK will then represent the distance of the valve from mid-position when cut-off takes place. This distance we know is the outside lap. Draw the arc KI, known as the lap circle, and it will cut the valve circle again at V. When the valve is again the distance OV = the outside lap from mid-position, admission will take place. Draw the line OVA and this will represent the position of the crank at admission.

When the crank is at OX, the valve displacement is equal to OJ. This is at dead center and the valve is open the amount IJ, for it has moved this distance more than the outside lap. Therefore IJ is the lead for this end.

Now on the other valve circle, draw the arc PQ with the inside lap ($\frac{3}{4}$ inch) as a radius. It will cut the valve circle at P and Q. When the valve displacement is equal to OQ, the exhaust port has just closed and the engine is at compression. In the same way OP is the valve displacement at release when the port begins to open. OQD represents the crank position at compression and OPB the crank position at release.

The results then are as follows:

Given:

Valve travel	= XY = 3 inches.
Angular advance	= angle MOE = 35°.
Inside lap	= OP = $\frac{3}{4}$ inch.
Crank angle at cut-off	= angle XOC 115°

Found:

Outside lap	= OK	= $\frac{3}{4}$ inch.
Angle of lead	= XOA	= 5°.
Linear lead	= IJ	= $\frac{3}{32}$ inch.
Max. port opening for admission	= HE	= $\frac{3}{4}$ inch.
Crank angle at compression	= XOD	= 185°
Crank angle at release	= XOB	= 65°
Max. port opening for exhaust	= FN	= $\frac{3}{4}$ inch.

Fig. 26 is drawn full size, and all of these measurements may readily be verified. This figure is drawn for the head end only. If the crank angle at cut-off is the same on both ends, the Zeuner's diagram for the crank end will be exactly like Fig. 26.

ANOTHER PROBLEM.

Given:

The valve travel	= 3 inches.
The lead angle	= 6°.
Crank angle at cut-off	= 70°.
Crank angle at compression	= 75°.

To Find:

Angular advance.
Laps.
Linear lead.
Crank angle at release.

As before, let XY represent the valve travel = 3 inches and draw OM and the circle XMYF. See Fig. 27. Lay off the lead

angle XOA = 6°. Then OA represents the crank position at admission. Next lay off the crank angle XOC, the angle at cut-off 70°. Bisect the angle COA by the line OE and on OE draw the valve circle. Angle MOE = the angular advance. The valve circle will cut the crank lines OC and OA at K and V respectively. If the work has been carefully done, OK will be exactly equal to OV and will represent the outside lap. The lead is IJ as before. Draw OD at position of the crank at compression so that angle XOD = 75°. Continue OE to cut the eccentric circle at F. On

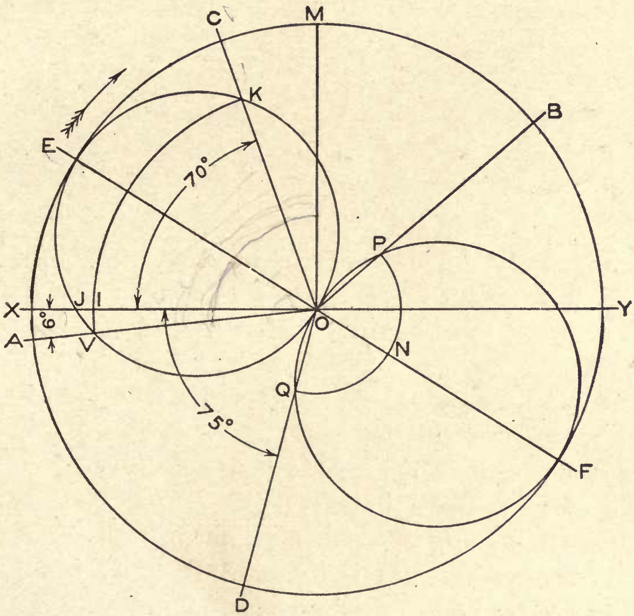

Fig. 27.

OF draw the second valve circle. It will cut OD at Q, and OQ will represent the inside lap. Draw the lap circle OP, and the crank position OPB. This will be the crank position at release.

The angular advance in this problem is large and all the events of the stroke are early. Compression and release are excessively early and the outside lap is unusually large. In the previous problem, with cut-off at about two-thirds stroke, the results were nearly normal. Cut-off with the plain slide valve, earlier than half stroke cannot be had without sacrificing the steam distribution on the other events.

To sum up we have

Given:

Valve travel	= XY	= 3 inches.
Lead angle	= XOA	= 6°.
Crank angle at cut-off	= XOC	= 70°.
Crank angle at compression	= XOD	= 75°.

To find:

Angular advance	= MOE	= 58°.
Outside lap	= OV	= $1\frac{5}{32}$ inches.
Lead	= IJ	= $\frac{3}{32}$ inch.
Inside lap	= OQ	= $\frac{7}{32}$ inch.
Crank angle at release	= XOB	= 130°.

Suppose in this last problem the cut-off had been given at half stroke instead of having the crank angle given, and that the compression had been given in the same way. We should, of course, need to know the ratio of length of connecting rod to crank. Let this be given as 4, that is, the connecting rod is four times the length of the crank.

In Fig. 28 let XY represent the valve travel. Extend XY to the left to the point Z, and make OZ equal to four times OX. With Z as a center and OZ as a radius, strike an arc OC that will cut the eccentric circle at C; then draw OC, which will represent the crank when the piston is at half-stroke, which is assumed to be the point of cut-off.

To find the crank angle at compression, lay off YH equal to .8 of the distance YX. From H lay off HW = OZ = four times OX. From W as a center with a radius WH, draw an arc cutting the eccentric circle at D. Draw OD, which will represent the position of the crank at compression.

The student is advised to read over again pages 13 to 14 if this explanation of finding the crank angle does not seem perfectly clear.

ANOTHER PROBLEM.

Given:

Cut off at .6 stroke.	
Lead	= $\frac{1}{16}$ inch.
Maximum port opening	= ½ inch.
Ratio of crank to connecting rod	= 4.

To find:

The eccentricity.
Lead angle.
Angular advance.
Laps.

In Fig. 29 assume an eccentricity that will if possible be a little too large. Let us take for trial $2\frac{1}{8}$ inches and draw XY equal to twice the assumed eccentricity equal to $4\frac{1}{4}$ inches. Lay off XC′ equal to .6 of XY, and with a radius equal to four times OX draw the arc C′C as already explained.

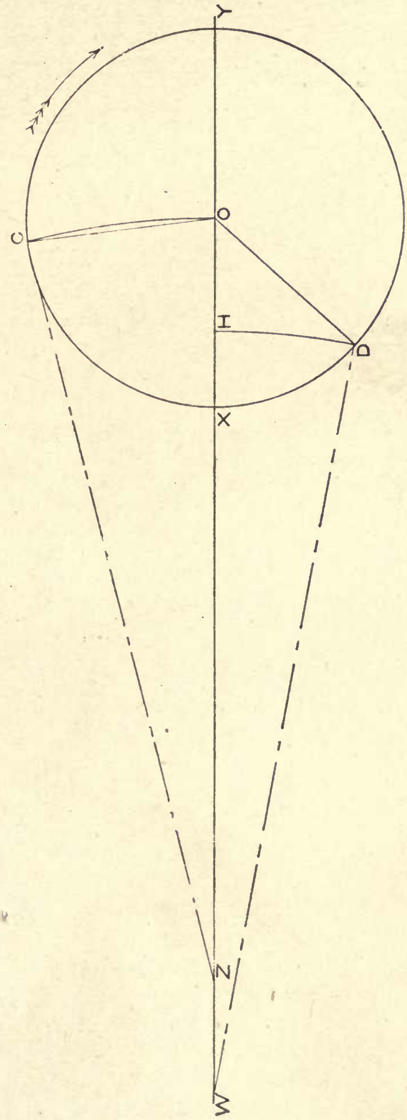

Fig. 28.

Then draw OC, which will represent the position of the crank at cut-off, and XOC will be the crank-angle at cut-off. Assume a lead angle of about 7° and draw OA, which, if this assumption be true, will represent the crank-angle at admission. Bisect the angle COA by the line OE, and on OE draw the valve circle. Draw the lap circle VNK. With this assumed eccentricity we find a maximum port opening of NE = .75 inch, which is larger than the conditions of the problem demand. We may then form a proportion, namely:

The actual port opening desired : the port opening with the assumed eccentricity : : probable eccentricity : assumed eccentricity.

Substituting the figures we have $.5 : .75 :: x : 2\frac{1}{8}$ ∴ x = the probable eccentricity; equals 1.42 inches.

Now draw on OE, a new valve circle (dotted) with a diameter equal to the required eccentricity of 1.42 inches. See Fig. 29*a*. It will cut the crank line

OC at K′, and OK′ will be the new outside lap and I′J′ will be the new lead (assuming the lead angle to be 7°). This lead I′J′ is $\frac{1}{8}$ inch, while the required lead is only $\frac{1}{16}$ inch. Now decrease the angular advance enough to correct one half of this difference, by drawing a new lap circle J″K″ of $\frac{1}{16}$ inch greater radius. This will make the valve circle cut OC at K″, so that OK″ will now be

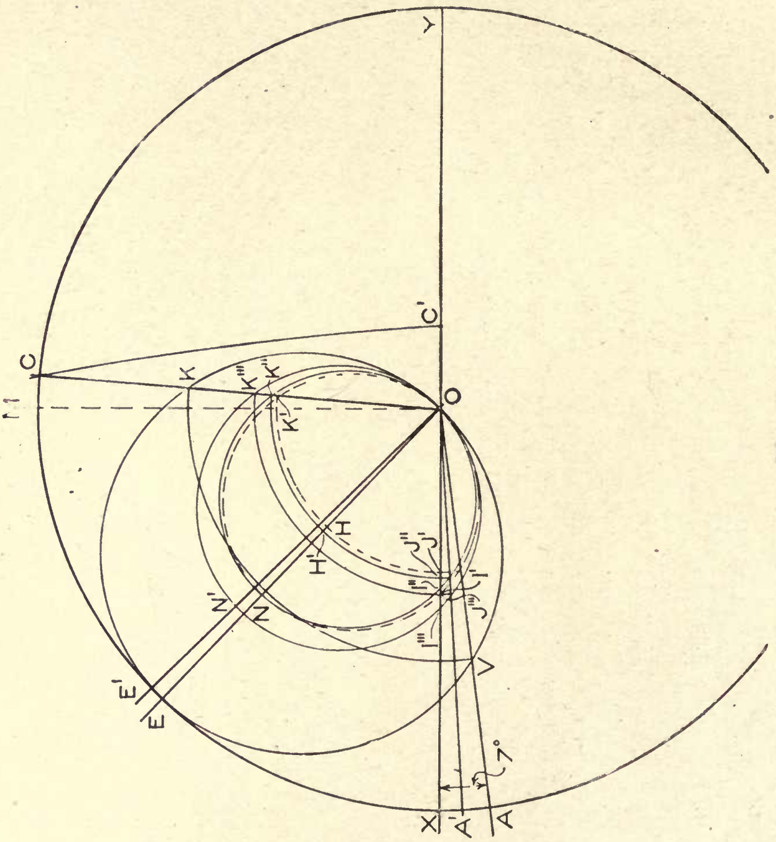

Fig. 29.

the final lap, and I″J″ the final lead, which is equal to the required $\frac{1}{16}$ inch. The lead angle is now XOA′ instead of XOA. The port opening at NH′ is $\frac{1}{2}$ inch (nearly) as required, but the change in angular advance necessitates an increase of lap if cut-off is to remain the same. This reduces the port opening by the amount HH′, so that the maximum opening is only .46 inch. By increasing the eccentricity this port opening may be increased.

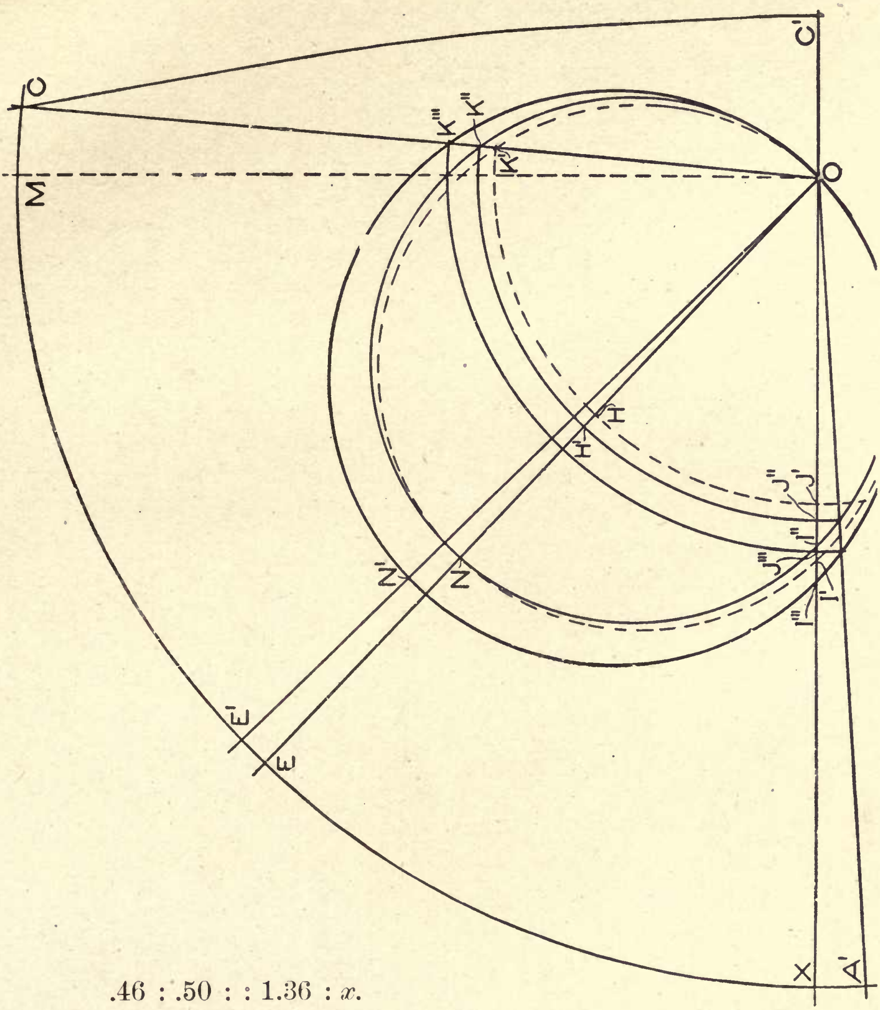

Fig. 29a. Essentials of Fig. 29 to larger scale.

$.46 : .50 :: 1.36 : x.$

$x = 1.48$, the true eccentricity.

Now draw the valve circle on OE′ with a diameter of 1.48 inches. It will cut OC in K‴ and OX in J‴. The lap will be OK‴ = .97 inch, the lead will be I‴ J‴ = $\frac{1}{16}$ inch, the angular advance will be MOE′ and the eccentricity ON′.

To sum up we have

Given:

The cut-off = .6 stroke.
The lead = $\frac{1}{16}$ inch
Max. port opening = ½ inch.

Obtained all of the above conditions together with:

Lap	= .97 inch.
Lead angle	= XOA′°
Angular advance	= MOE′°

Compression, release and inside laps are found as in the previous problems.

There are of course all sorts of combinations that would make up different problems, but they can all be solved in the same general way, as they are modifications of the above.

DESIGN OF THE SLIDE VALVE.

In designing a slide valve some of the variables are assumed and the others are found by means of diagrams as we have already seen. These diagrams show only the dimensions of the inside and outside laps and travel of valve; the other dimensions of the valve and seat must be calculated.

Area of Steam Pipe. Pipes that supply the steam chest should be large enough to prevent an excessive loss of pressure due to friction. If the pipes are long they should be of such size that the mean velocity of steam in them does not exceed 100 feet per second or 6,000 feet per minute. For this calculation it is usual to assume steam admitted to the cylinder during the whole stroke.

For example. Suppose an engine is 10″ × 18″, and makes 180 revolutions per minute. What is the diameter of the steam pipe?

The piston displacement or volume of the cylinder is:

$$\frac{\pi d^2}{4} \times l = \frac{3.1416 \times 10^2}{4} \times 18 = 1413.72 \text{ cubic inches.}$$

$$\frac{1413.72}{1728} = .818 \text{ cubic feet.}$$

If the engine makes 180 revolutions it would use $2 \times 180 \times .818 = 294.48$ cubic feet per minute.

The area would be $= \frac{294.48}{6000} = .04908$ sq. ft. $= 7.0675$ sq. in.

The diameter corresponding to 7.0675 square inches is 3 inches.

A three-inch pipe would be large enough, especially if the engine cut off early in the stroke.

For a very large engine cutting off early, the allowable velocity may be taken as 8,000 feet per minute instead of 6,000 feet.

Width of Steam Port. The port opening at admission should give nearly as great an area as the steam pipe in order to prevent loss of pressure due to wire-drawing, but the actual width of the port should be great enough for the free exhaust of steam. It is well to have the steam port a little larger than the area of the steam pipe, then with a port opening of .6 to .9 of the port area for admission and full port opening at exhaust, satisfactory conditions will result.

The length of the ports is usually made about .8 the diameter of the cylinder. Then in the 10" × 18" engine the steam ports would be 8 inches long. If the area for admitting steam is 8.0675 square inches and the length of port is 8 inches, the width will be

$$= \frac{7.0675}{8} = .8834 \text{ inch, or about } \tfrac{7}{8} \text{ inch.}$$

The width of port opening would be about $.9 \times .8834 = .79506$ inch or about $\frac{13}{16}$ inch.

Width of Exhaust Port. When the slide valve is at its maximum displacement, the valve overlapping the exhaust port as shown in Fig. 7 reduces the area more or less. In designing the valve, the exhaust port should be of such a width that the maximum displacement of the valve does not reduce the area of the exhaust port to less than the area of the steam port. It is not advisable to make the exhaust port too large for this increases the size of the valve and thus causes excessive friction.

The height of the exhaust cavity should never be less than the width of the steam port, and may be made much higher to advantage.

Width of Bridge. The bridge must be of sufficient width so that outside edges of the valve cannot uncover the exhaust port. The width of the steam port plus the width of the outside lap plus the width of the bridge must be greater than the maximum displacement.

The width of the bridges should be not less than the thickness of the cylinder wall in order to make a good casting.

The Point of Cut-off. In the study of Indicators, it was shown that if the point of cut-off is early, the other events are not good. If a plain slide valve is used with an automatic cut-off, the cut-off

is hastened either by changing the eccentricity or by changing the angular advance. Either of these methods will accomplish the result at the expense of the compression which consequently will be earlier and excessive. Except for locomotives and high-speed engines, where compression is an advantage, the plain slide valve is not arranged to cut-off earlier than $\frac{1}{2}$ or $\frac{5}{8}$ stroke. If an earlier cut-off is desired, large outside laps are necessary. The cut-offs may be equalized by giving the head end a greater lap than the crank end, but this will cause an inequality of lead.

Lead. The lead of stationary engines varies from zero to $\frac{3}{8}$ inch according to the style of engine. An engine having high compression that compresses the steam nearly to boiler pressure, will give good results with little or no lead. If the ports are small, and the clearance large, there should be considerable lead in order to insure full initial pressure on the piston at the beginning of the stroke. Valves that open slowly require more lead than quick-acting valves.

Let us design and lay out the valve and valve seat for the following engine:

Diameter of cylinder = 10 inches.
Stroke = 18 inches.
Revolutions = 180 per minute.
Lead angle = 3°.
Cut-off to be equal at both ends and to take place at .75 stroke.
Max. port opening = .9 area of steam pipe.
Compression to be .85 of the stroke at both ends.
Length of connecting rod = 3 feet.

The piston displacement, or cylinder volume, will be $\frac{3.1416 \times 10^2}{4} \times 18 = 1413.7$ cubic inches or .818 cubic feet. If the engine makes 180 revolutions, it will use $2 \times 180 \times .818 = 294.48$ cubic feet of steam per minute. Steam pipe area $= \frac{294.48}{6000} = .0491$ square feet = 7.07 square inches.

This 7.07 square inches would also be the least possible area of the steam ports. If the length of port is made .8 the diameter of cylinder, the width will be $\frac{7.07}{8} = .88$ inches or about $\frac{7}{8}$ inch. The width of maximum port opening will be $.9 \times .88 = .792$ or nearly $\frac{13}{16}$ inch.

It will be necessary to draw a separate valve circle for each end of the cylinder. First consider the head end.

The valve travel not being known, we shall lay off XY on an assumption of 6 inches travel and draw the eccentric circle as shown in Fig. 30. Lay off the lead angle XOA = 3°. Lay off XC′ = .75 of the assumed valve travel = $4\frac{1}{2}$ inches. Draw the arc CC′ as previously explained and draw OC which will be the crank angle at cut-off. The radius of the arc C′C will be equal to

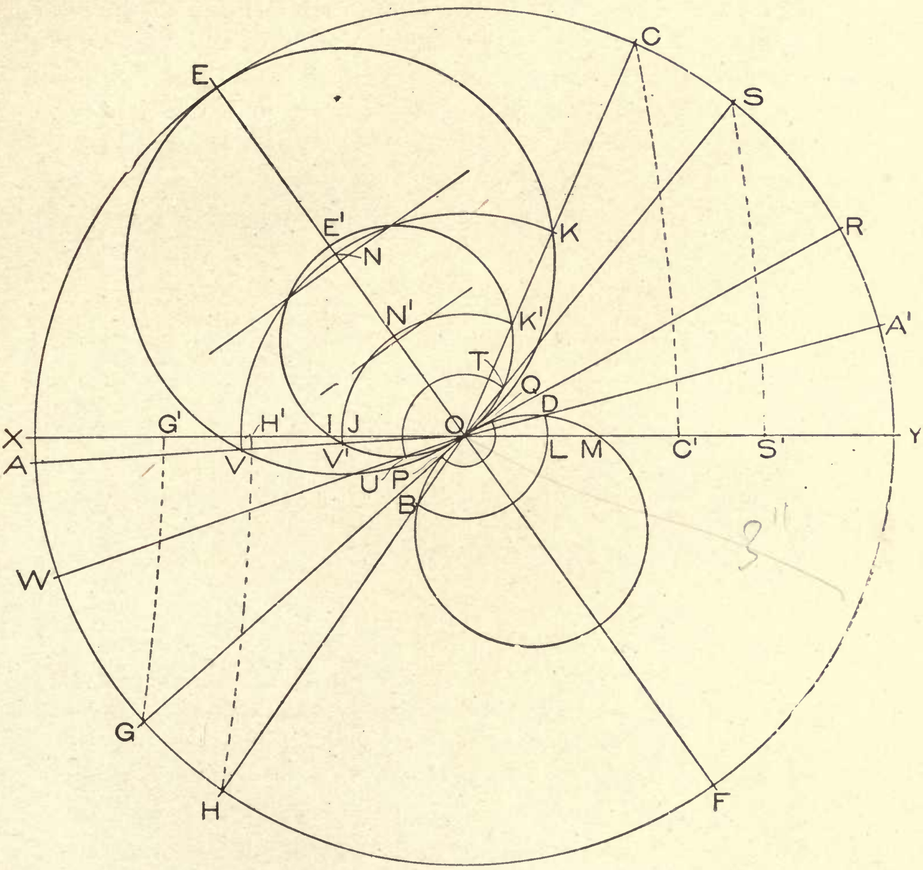

Fig. 30.

4 times the radius of the eccentric circle, or 12 inches, because the connecting rod is 4 times the length of the crank. Bisect the angle AOC by the line OE, and on OE draw the valve circle. OV = OK is then the outside lap, with these assumed condi-

tions. Draw the lap circle; then EN will be the maximum port opening. EN $= 1\frac{7}{16}$ inches, while $\frac{13}{16}$ inch is all that is necessary. The assumed eccentricity is 3 inches, therefore the probable eccentricity $= x : 3 :: \frac{13}{16} : 1\frac{7}{16}$. $x = 1\frac{11}{16}$ inches.

Now draw a new eccentric circle with a radius of $1\frac{11}{16}$ inches and a new valve circle with OE′ $= 1\frac{11}{16}$ inches as a diameter. OK′ is now the outside lap and the maximum port opening is equal to E′N′, which from actual measurement is found to be $\frac{13}{16}$ inch. The outside lap = OK′ = OV′ = $\frac{27}{32}$ inch and the lead is IJ = $\frac{3}{32}$ inch.

Produce EO to F and draw another valve circle. We shall use this valve circle to determine the outside laps and lead for the crank end of the cylinder. Since the cut-off is to be .75 of the stroke, we may lay off OH′ = OC′, and with a radius of 12 inches

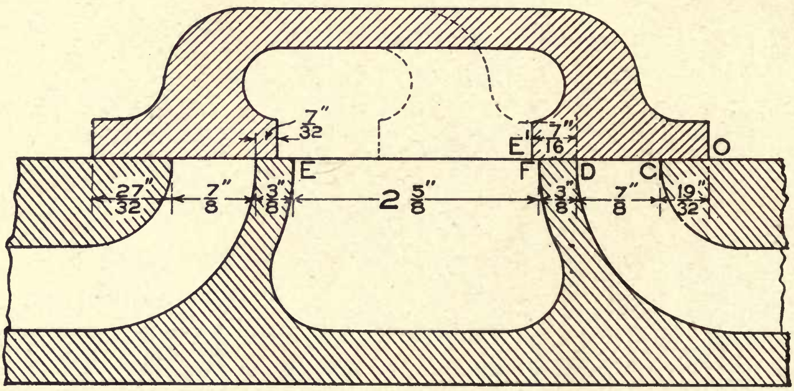

Fig. 31.

draw the arc HH′. Then, as already explained, OH will be the crank angle at cut-off on the return stroke. OB will be the outside lap $= \frac{19}{32}$ inch. Draw the lap circle intersecting the valve circle at D. Then ODA′ is the crank angle at admission on the return stroke and LM $= \frac{3}{8}$ inch is the lead on the crank end of the cylinder. The maximum port opening will always be greater at the crank end than at the head end because the crank end lap is less in order to get the equal cut-off. If the laps were equal, of course the port openings would be equal.

Now lay off YG′ = .85 of XY and find the crank position OG. This is the compression on the head end of the cylinder and gives an inside lap on this end of $\frac{7}{32}$ inch, which is equal to OP.

Draw the lap circle PQ, which allows us to draw through Q the crank line OR, which is the release on the forward stroke.

Lay off XS′ = YG′ = .85 of XY, and construct the crank line OS, which is the crank position at the crank end compression. OS intersects the valve circle at T, which gives OT = $\frac{7}{16}$ inch = inside lap on the crank end. Draw this lap circle, which will intersect the valve circle at U. This enables us to draw OUW, the crank angle at release, on the return stroke.

From the data determined by means of these diagrams the valve may now be laid out. For convenience let us tabulate the results obtained as follows:

Data.	Head End.	Crank End.
Cut-off, per cent of Stroke	75	75
Outside Lap	$\frac{27}{32}''$	$\frac{19}{32}''$
Inside Lap	$\frac{7}{32}''$	$\frac{7}{16}''$
Lead	$\frac{3}{32}$	$\frac{3}{8}''$
Port Opening	$\frac{13}{16}''$	$1\frac{1}{16}''$
Width of Port	$\frac{7}{8}''$	$\frac{7}{8}''$

Fig. 31 shows this valve in section. Let us begin at the end having the largest inside lap, or in this case at the crank end. Lay out the steam port $\frac{7}{8}$ inch wide, and the crank-end outside lap = $\frac{19}{32}$ inch. The bridge will be, say, $\frac{3}{8}$ inch wide. From the inner edge of the steam port, lay off the crank-end inside lap = $\frac{7}{16}$ inch. When the valve moves to the left, the point E′ will travel $1\frac{1}{16}$ inches, a distance equal to the eccentricity, and in this position of extreme displacement the exhaust port EF must be open an amount at least equal to the steam port, $\frac{7}{8}$ inch. Therefore we lay off EF equal to $1\frac{1}{16}'' + \frac{7}{8}'' = 2\frac{9}{16}''$. The inside lap overlaps the bridge nearly $\frac{1}{8}$ inch, so that we shall have to make the exhaust port opening equal to $2\frac{5}{8}$ inches. Lay off $\frac{3}{8}$ inch again for the bridge and measure back $\frac{7}{32}$ inch, equal to the head-end inside lap. The port is $\frac{7}{8}$ inch wide, and the head-end inside lap of $\frac{27}{32}$ inch completes the outline of the valve seat.

VALVE SETTING.

The principles of valve diagrams are useful in setting valves as well as in designing them. The valve is usually set as accurately as possible, and then, after indicator cards have been taken,

the final adjustment can be made to correct slight irregularities.

The slide valve is so designed that the laps cannot be altered without considerable labor, and the radius of the eccentric, which determines the travel of the valve, is usually fixed. The adjustable parts are commonly the length of the valve spindle and the angular advance of the eccentric.

By lengthening or shortening the valve spindle, the valve is made to travel an equal distance each side of the mid-position. Moving the eccentric on the shaft makes the action of the valve earlier or later as the angular advance is increased or decreased.

To Put the Engine on the Center. It is usual to put the engine on center before setting the valve. First put the engine in a position where the piston has nearly completed the outward

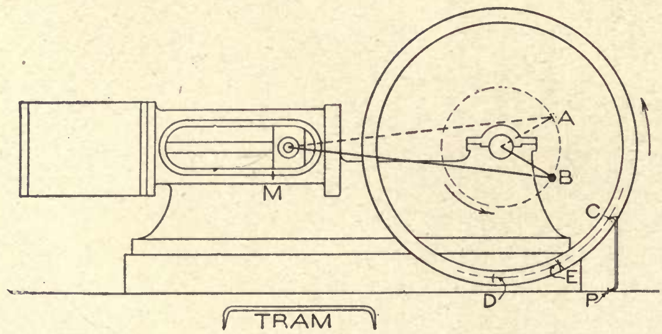

Fig. 32.

stroke, and make a mark M on the guide opposite the corner of the crosshead or at some convenient place. Also make a mark, with a center punch, on the frame of the engine near the crank disc or on the floor. With this punch mark P as a center, describe an arc C on the wheel rim, with a tram. A tram is a steel rod with its ends bent at right angles and sharpened.

Turn the engine past the center until the mark on the guide again corresponds with the corner of the crosshead, and make another mark D on the wheel with the tram, keeping the same center. With the center of the pulley or crank disc as a center, describe an arc CD on the rim, which intersects the two arcs drawn with the tram. Bisect the arc CD on the rim, included between the two short arcs, and turn the engine until the new point E is at

a distance from the point on the frame equal to the length of the tram, in which position the engine will be on the center.

The engine should always be moved in the direction in which it is to run so that the lost motion of the wrist pin and crank pin will be taken up the right way. In case the engine has been moved too far at any time, it should be turned back beyond the desired point and brought up to that point while the engine is moving the right way.

To Set the Valve with Equal Lead. Set the engine on the dead point and give the eccentric the proper angular advance. Adjust the length of the valve spindle to give the proper lead for that end. Now place the engine on the other dead point and measure the lead at that end. If the leads are unequal, correct half the error by changing the length of the valve spindle and the other half by altering the angular advance. In case the valve gear has a rocker, the length of the spindle should be such that the rocker will move as designed. The angular advance should not be changed, but the equal lead should be obtained by means of the valve spindle or the eccentric rod.

Second Method. In case it is difficult to turn an engine the following method may be used. First loosen the eccentric on the shaft and turn it around until it gives maximum port opening first at one end and then at the other. If the maximum port openings are not equal, make them so by changing the length of the valve spindle by half the difference. When the above adjustment has been made, set the engine on dead center and give the valve the proper lead by turning the eccentric on the shaft. The angular advance is thus adjusted.

To Set the Valve for Equal Cut-off. Place the engine on the dead point, give the eccentric the proper angular advance and the valve the proper lead. Move the engine forward until cut-off occurs, then measure the displacement of the crosshead from the beginning of the stroke. Continue moving the engine forward, until cut-off takes place on the return stroke and measure the displacement of the crosshead from the beginning of this stroke to this point.

In case the cut-off is earlier at the crank than at the head-end, the valve spindle is too short. Adjust the length of the spindle

so that the inequality will be corrected. Now set the engine on the dead point again and give the valve the proper lead by means of the eccentric. By repeating the process, making slight changes, the desired result will be obtained.

MODIFICATIONS OF THE SLIDE VALVE.

The ordinary slide valve is suitable for small engines; but for large sizes some method must be employed to balance the steam pressure on the back of the valve. With large valves, such for instance as those of locomotives or large marine engines, a great force is exerted by the steam, and the valve is forced against its seat so hard that a large amount of power is necessary to move it. This excessive pressure causes the valve to wear badly and is a dead loss to the engine. The larger the valve, the greater this loss will be.

Piston Valve. To prevent excessive pressure on the back of the valve, the piston valve is commonly used, especially in marine engines. This valve consists of two pistons, which cover and uncover the ports in precisely the same manner as the laps of the plain slide valve. These pistons are secured to the valve stem in an approved manner and are fitted with packing rings.

The valve seat consists of two short cylinders or tubes accurately bored to fit the pistons of the valve. The port openings are not continuous as in the plain slide valve, but consist of many small openings, the bars of metal between these openings preventing the packing rings from springing out into the ports.

Steam may be admitted to the middle of the steam chest and exhausted from the ends or vice versa. With the former method, the live steam is well separated from the exhaust, and the valve-rod stuffing box is exposed to exhaust steam only. This is a good arrangement for the high-pressure cylinder; if used for a cylinder in which there is a vacuum, air may leak into the exhaust space through the valve-rod stuffing box. With this arrangement the steam laps must be inside and the exhaust laps on the outside ends.

The piston valve may be laid out and designed by means of the Zeuner diagram just as if it were a plain slide valve, and the action is the same except that it is balanced so far as the steam

pressure is concerned; the power to drive it being only that necessary to overcome the friction due to the spring rings.

Fig. 33 shows a section of the piston valve and the high-pressure cylinder for one of the engines of the U. S. S. "Massachusetts." This valve consists of two pistons connected by a sleeve through which the valve rod passes. This valve rod is prolonged to a small balancing piston, placed directly over the main

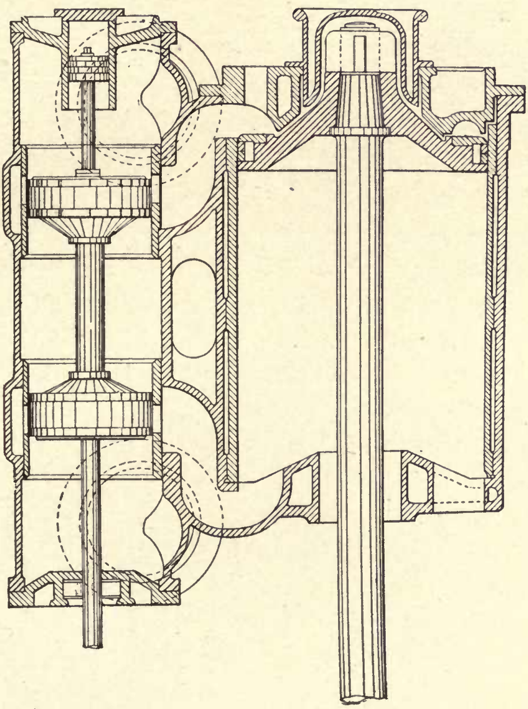

Fig. 33.

valve. The upper end of the balancing cylinder does not admit steam, so that the steam pressure below the balancing piston will practically carry the weight of the piston valve, thus relieving the valve gear and making the balance more nearly complete.

Double-Ported Valve. Sometimes it is impossible to get sufficient port opening for engines of large diameter and short stroke, especially those having a plain slide valve with short travel.

This difficulty may be overcome by means of the double-ported valve shown in Fig. 34. It is equivalent to two plain slide valves, each having its laps. The inner valve is similar to a plain slide valve except that there is communication between the exhaust space and the exhaust space of the outer valve. Each passage to the cylinder has two ports; a bridge separates the exhaust of the outer valve from the steam space of the inner valve, and the outer valve is made long enough to admit steam to the inner valve.

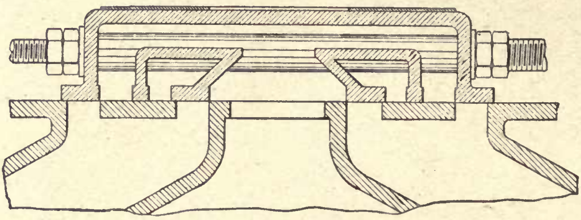

Fig. 34.

This valve may be considered as equivalent to two equal slide valves of the same travel, each having one-half the total port opening. To admit the same amount of steam as a plain slide valve, the double-ported valve requires but half the valve travel; this is advantageous in high-speed engines.

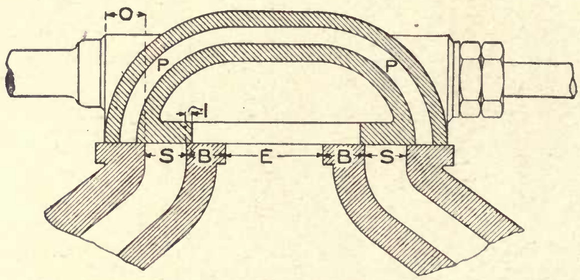

Fig. 35.

To balance the excessive steam pressure, the back of the valve is sometimes provided with a projecting ring which is fitted to a similar ring within the top of the valve chest. These rings are planed true, and fit so that steam is prevented from acting on the back of the valve. The space inside the rings is sometimes placed in communication with the condenser.

The Trick Valve. The defect of the plain slide valve, due to the slowness in opening and closing, is largely remedied in the trick valve, which is so made that a double volume of steam enters during admission. Thus a quick and full opening of the port is obtained with a small valve travel.

In Fig. 35 the valve is shown in mid-position. It is similar to a plain slide valve except that there is a passage PP through it. It has an outside lap O and an inside lap I. The seat is raised and has steam ports SS, bridges BB, and exhaust port E. If the valve moves to the right a distance equal to the outside lap plus the lead, it will be in the position shown in Fig. 36. Steam will be admitted at the extreme left edge of the valve just the same as though it were a plain slide valve; also, since steam surrounds the valve it will be admitted through the passage as shown in Fig. 36.

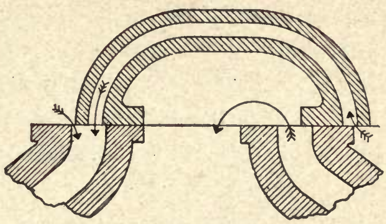

Fig. 36.

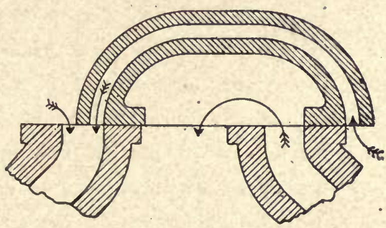

Fig. 37.

If the lead is the same as for a plain slide valve, $\frac{1}{16}$ inch for instance, this valve would give double the port opening, that is $\frac{1}{8}$ inch, when the valve was open a distance equal to the lead.

Fig. 37 shows the valve when it is in its extreme position to the right and the port is full open to steam.

Piston valves are also made with a passage similar to that of the trick valve for double admission. The valve used with the Armington and Sims engine is perhaps the best example.

Balanced Valves. Since there is a wide difference between the pressure of admission and exhaust, there must always be a great pressure acting upon the valve, causing it to run hard and wear excessively. The greater the steam pressure, the lower the pressure at exhaust and the larger the valve, the greater this pressure will be.

Piston valves are commonly used on the high and intermediate cylinders of triple-expansion engines, and if well made and fitted with spring rings, should not leak. Small piston valves are often made without packing rings; but even if they fit accurately when new, they soon become worn and cause trouble.

The double-ported valve, the trick valve, and others often have some device for relieving the pressure, such as a bronze ring or cylinder, fastened to the back of the valve. This ring is pressed by springs against a finished surface of the valve chest cover, and the space thus enclosed by the ring may be connected to the exhaust. There are numerous devices for balancing valves, but they are usually more or less expensive and are liable to cause trouble from leakage.

STEPHENSON LINK MOTION.

One of the earliest, and at present one of the most common mechanisms for reversing engines, or changing the ratio of expansion, is the Stephenson link motion, shown in Fig. 38. This illustration is taken from the drawings of a recent battleship engine, and may be considered the typical arrangement of the Stephenson gear as applied to marine practice.

The two eccentrics E and E′, whose centers are at C and C, respectively, are shown in their relative positions when the crank OA is at dead center. The eccentric rods R and R′ are connected by forked ends to the link pins H and G. The link consists of two curved bars bolted together in such a manner that they may slide by the link block N. On the link are three sets of trunions; the two outer ones, or link pins, are fitted into the forked end of the eccentric rods, and the middle one, known as the saddle pin, is fitted into the end of the drag links FM.

The valve stem has, at its lower end, a pivoted block N, called the link block, provided with slotted sides through which the links can slide from right to left. The reverse shaft, or rock shaft, K, here shown in full gear "forward," may be turned until F moves over to B; in this position the link will be pushed across the link block, and the valve will get its motion from the rod R′ instead of from R as before. The link in this position would be full gear "astern"

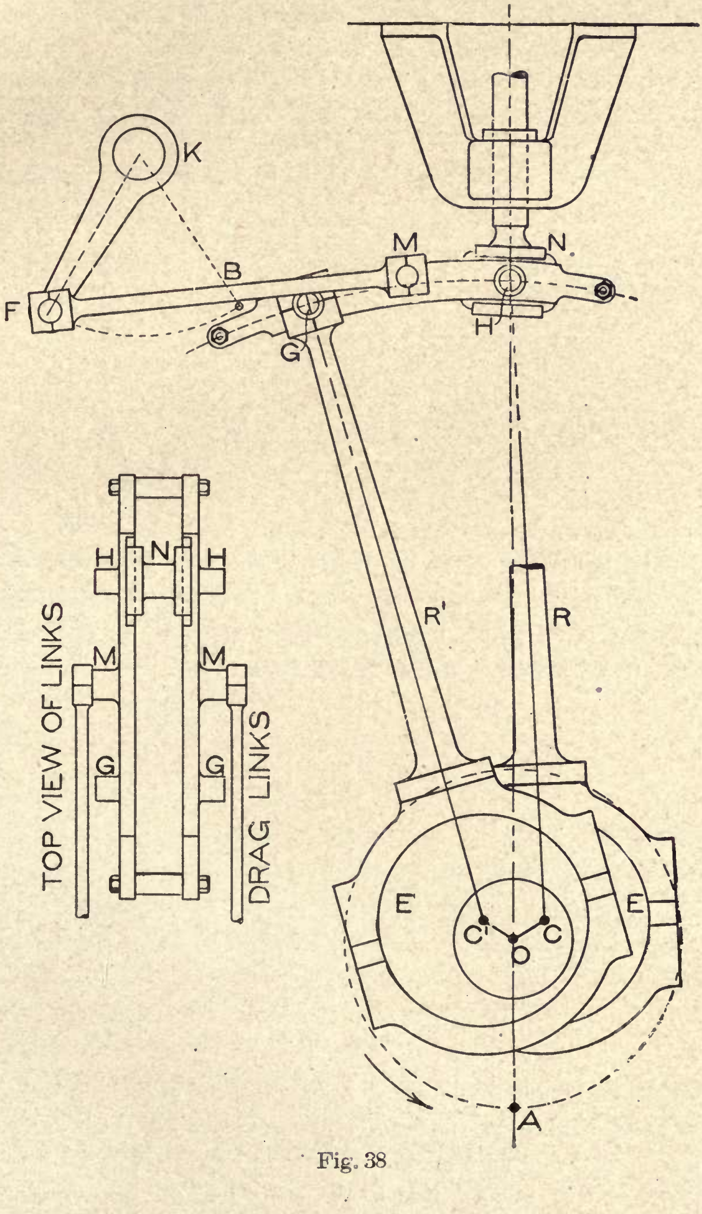

Fig. 38

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In all large engines, such as marine, the reverse shaft is turned by power, but in smaller engines, such as locomotives, the engineer can turn the shaft by means of a lever.

When set full gear forward, as in Fig. 38, the valve admits steam to the crank end of the cylinder, and the crank revolves as

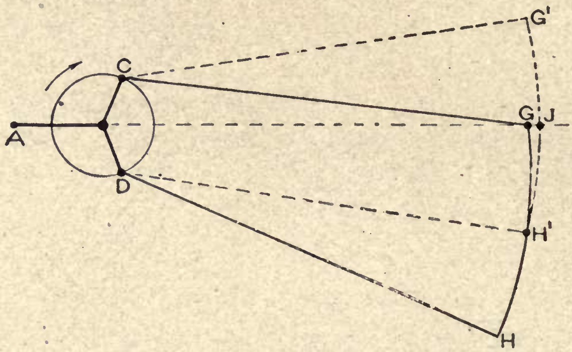

Fig. 39.

shown by the arrow. As the crank turns, both eccentrics impart motion to the link, but the "go ahead" link pin H approximately coincides with the link block, so that nearly all its up-and-down motion is transmitted to the valve stem, while the "go astern"

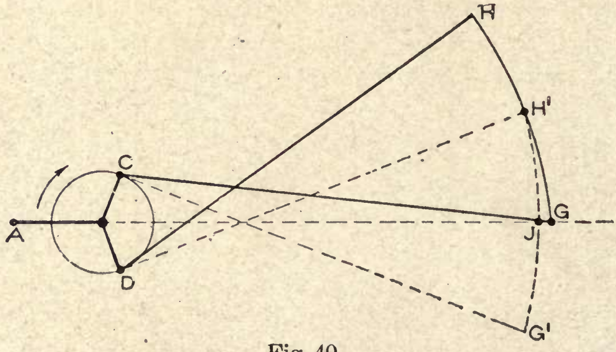

Fig. 40.

eccentric exerts but little effect upon the link block. Moving the drag links over to the extreme right reverses all these conditions by bringing the other link pin under the link block. In this position, steam will be admitted to the other end of the cylinder, and the engine will run in the opposite direction. This will be clearly seen by referring to Fig. 38.

When at full gear, either forward or backing, the valve moves as if there were really but one eccentric, while at intermediate points its motion is the result of the combined influence of both eccentrics, one tending in a measure to counteract the other. The effect of this is to shorten the valve travel the same as if the valve were driven by a new eccentric having less throw than either of the other two.

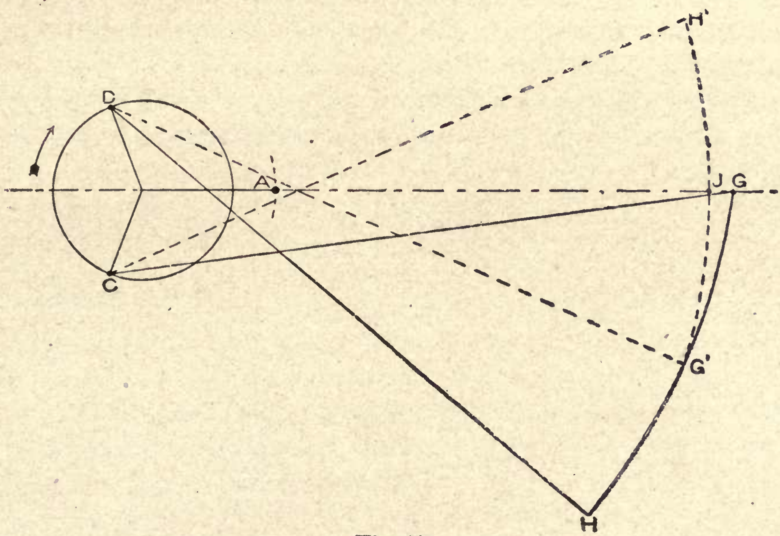

Fig. 41.

Decreasing the valve travel causes cut-off to occur earlier compression is earlier, release later, and the lead is reduced somewhat. If every point of the link moved in the arc of a circle when the drag link is shifted, the lead would not alter; but, since the eccentric rods about which each end swings are centered at different points, C and C′, this is impossible.

Figs. 39 and 40 show the two principal ways of arranging the eccentric rods of a Stephenson gear. The first is said to have "open rods", the second "crossed rods"; referring to whether the rods are crossed or open when both the eccentrics face the link. It can easily be seen that when the eccentrics shown in Fig. 39 have turned through 180° they will be in the position shown in Fig. 41, but this is the same arrangement as before and is "open" rods. The full lines show the positions in full gear forward, while

the dotted lines indicate the positions in mid gear. With open rods it will be seen that when at full gear the link block is at G, and that if, without turning the crank, the link is shifted to mid gear, then the link block moves to J, Fig. 39, and the valve must consequently be moved toward the right an amount equal to GJ, thereby increasing the lead on the crank end of the cylinder. With crossed rods, moving the link from full to mid gear moves the link block from G to J, Fig. 40, thus reducing the lead. It follows then that open rods give increasing lead from full toward mid gear, and that crossed rods give decreasing lead. With crossed rods there will be no lead when in mid gear. It will be apparent that the shorter the rods the greater this increase or decrease will be.

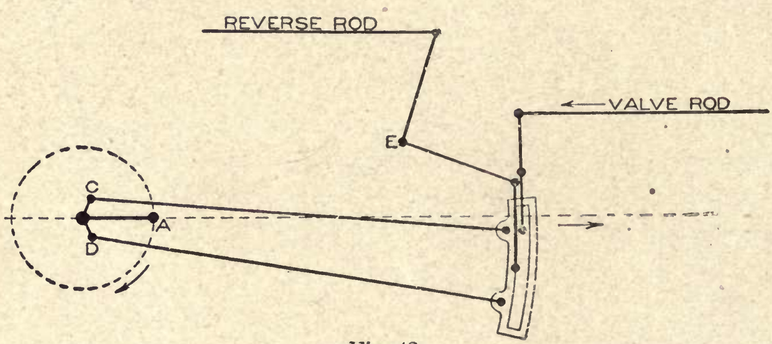

Fig. 42.

Nearly all marine engines, and some English locomotives, have their link blocks carried directly on the valve rod. American locomotives commonly use a rocker, one end of which carries the link block while the other moves the valve rod. This arrangement indicated in Fig. 42 makes it possible to place the valve and steam chest above the cylinder. The position of the crank for the same valve position is just opposite that shown in Fig. 39 because the rocker reverses the valve motion; this gives an arrangement of crank and eccentrics that is identical with that indicated in Fig. 41 and the rods, although apparently crossed, are in reality of the open rod arrangement, giving increasing lead toward mid gear. A rod from the bell-crank lever on the reverse shaft E, leads back to the engineer's cab and connects with the reverse lever. This ever moves over a notched arc, and may be held by a latch in any

one of the notches, thus setting the link in any position from mid gear to full gear, either forward or back.

The Stephenson link is designed to give equal lead at both ends of the cylinder; but to accomplish this, the radius of the link arc (that is an imaginary line in the center of the slot) must be equal to the distance from the center of this slot to the center of the eccentric. In Fig. 38 the radius of the link arc is equal to CH and C′G.

Exact quality of lead is not essential, and the radius of the link arc is sometimes made greater or less than stated above in order to aid in equalizing the cut-off; but the change should never be great enough to affect the leads.

Stephenson originally intended to use the link simply as a reversing gear, but soon found, however, that at intermediate points between the two positions of full gear, it would serve very well as a means of varying the expansion and cut-off. Very soon the link came to be used not only on locomotives and marine engines, but on stationary engines as well, in connection with the reverse shaft which was under the control of the governor. The mechanism proved to be too heavy to be easily moved by a governor and it has gradually fallen into disuse on stationary engines excepting as a means of reversing.

In marine practice, the variable expansion feature is of little value, for marine engines run under a steady load and the link is set either at full gear or at some fixed cut-off. For locomotives, however, the variable expansion is nearly as important as reversing. Locomotives are generally started at full gear, admitting steam for nearly the entire stroke, and then exhausting it at relatively high pressure. This wasteful use of steam is necessary to furnish the power needed in starting a train. After the train is under way, less power is required per stroke, and the link is gradually moved toward mid gear, or "notched up" by the engineer, thus hastening the cut-off; the expansion is increased and the power is reduced in proportion to the load.

As the cut-off is changed, it is desirable to maintain an approximately equal cut-off at each end of the cylinder; this can be secured in the Stephenson gear by properly locating the saddle pin and the reverse shaft. When used without a rocker, as in

Fig. 38, the saddle pin should be on the arc of the link or slightly ahead of it. When used with a rocker, the saddle pin should be behind the link arcs, and to give symmetrical action for forward and backward running, it should be opposite the middle of the arc, that is, equally distant from each link pin.

The Stephenson link cannot be designed directly from the Zeuner diagram, but a systematic investigation can be made by using a wooden model of the proposed link. This can be mounted on a drawing board, and the effect of changing the position of pins and the proportions of rods and levers can be determined without difficulty. By a system of trials a combination can be found best suited to obtain the desired results. Moreover, a model makes it possible to measure directly the slip cf the link block along the link. This slip should be kept as small as possible to prevent rapid wear. It can be controlled to some extent by properly locating the link pins, by avoiding too short a link, and by choosing a favorable position for the reverse shaft.

The Gocch Link. Another form of link motion, known as the Gooch Link, is illustrated in Fig. 43. It has been extensively used on European locomotives, although it is gradually being replaced by a type of valve gear known as the Walschaert, which will be described later.

The Gooch link has its concave side turned toward the valve instead of toward the eccentric. The radius of curvature of the link is equal to AB, the length of the radius rod. The link is stationary and the link block slides in the link. The engine is reversed by means of the bell-crank lever on the reverse shaft E which shifts the link block instead of the link, as is the case with the Stephenson. The link is suspended from its saddle pin M, which is connected by a rod to the fixed center F, so that the link can move forward and back as the eccentricity is changed, or it can pivot about its saddle pin as the eccentrics revolve.

Since the radius of the link arc is equal to AB, it is apparent that the block can be moved from one end of the link to the other, that is, from full gear "forward" to full gear "back" without moving the point A, which is on the end of the valve rod. The lead then is constant for all positions of the block, and the distribution of steam for locomotives is slightly preferable to that

obtained by the Stephenson; but the gear is more complicated and requires nearly double the distance between shaft and valve stem.

The variable lead is perhaps a slight advantage to the locomotive, which is a slow-speed engine in starting, thus requiring but little lead. As the speed increases, and the link is "notched up", the lead is increased as the cut-off is shortened, and at high speed we have a large lead. With the Gooch link, the lead can be set for the average running speed, and although a little too great for good work at slow speed, it is a matter of small consequence, because the engine runs at slow speed but a very small fraction of the time it is in service, and the loss due to large lead at slow speed is of no consequence whatever in a day's run.

Several other link motions have been used; but at the present time probably more Stephenson link motions are used than all

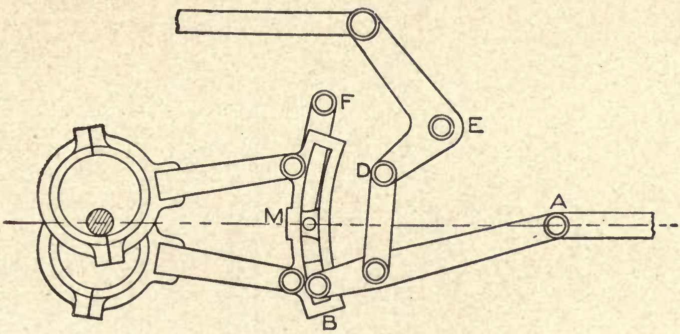

Fig. 43.

the other forms of reversing gear combined, and when a "link motion" is mentioned, the Stephenson is usually meant unless otherwise specified.

RADIAL VALVE GEARS.

In general, it would be desirable to have precisely similar steam distribution at each end of the cylinder, and it would often be of great advantage with an expansion gear like the Stephenson, if the cut-off could be shortened without changing any other event of the stroke. A Stephenson gear can be made to maintain equality of lead for both ends of the cylinder as the cut-off is shortened, but we have seen that in so doing, the lead of both ends is either

increased or diminished according as the link is arranged with "open rods" or "crossed rods". Moreover, the compression is hastened by bringing the link to mid-gear, all of which in many instances is undesirable.

This disadvantage of the Stephenson link motion lead to the design of the so-called "Radial Valve Gears", many of which are so complicated as to be impracticable, but all of which obtain a fairly uniform distribution of steam.

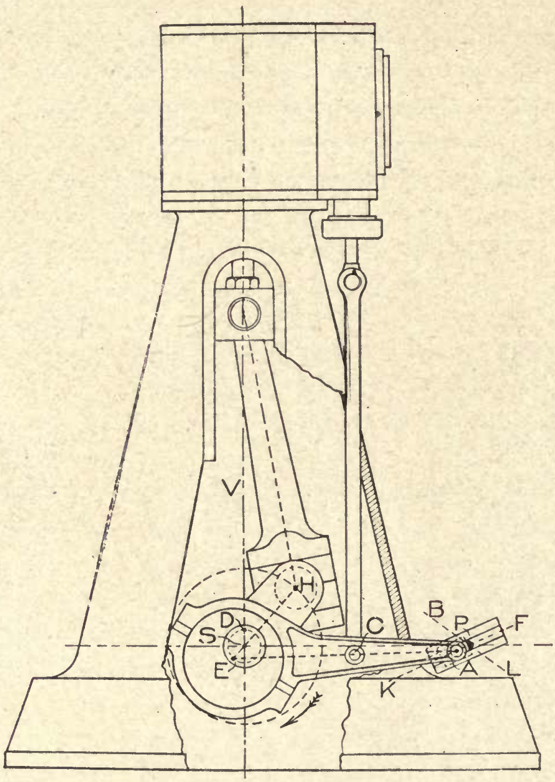

Fig. 44.

Hackworth Gear. The essential features of the Hackworth Gear are indicated in outline in Fig. 44. In this figure, S is the center of the shaft, and the eccentric E is set 180° from the crank SH. At the right-hand end of the eccentric rod EA, is pivoted a block which slides in a straight, slotted guide. The guide remains stationary while the engine is running, but can be turned on its

axis P, to reverse the engine or change the cut-off. P is a pivot, located on the horizontal through S in such a position that DP = EA. If these two distances are equal, A will coincide with P when the crank is at either dead point and the slotted guide may be turned from "full gear forward", as shown in the figure, through the horizontal position to "full gear backing", as

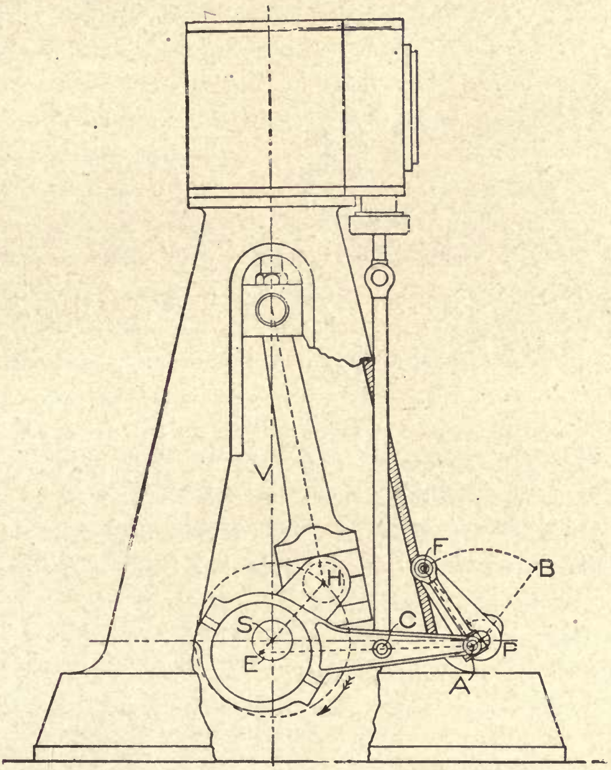

Fig. 45

shown by the line BL, without moving the valve. Therefore the leads are constant for all positions of the guide. The valve rod running upward from C, connects with the valve stem which it moves in a straight line. The valve stem is made just long enough to equalize both leads, and if the point C has been properly chosen, the two cut-offs will be very nearly equal for all grades of the gear.

A somewhat better valve action is obtained by slightly curving the slotted guide, with its convex side downward. This gear is sometimes used on marine engines and on small stationary engines.

Marshall Gear. The most objectionable feature of the Hackworth gear, is the slotted guide, for the sliding of the block causes considerable friction and wear. The Marshall gear, shown in outline in Fig. 45, is designed to obviate this feature. The point A moves in the desired path by swinging on the rod FA about F as a center. While the engine is running, the lever FP remains stationary, but can be turned on its axis P to reverse the engine, or change the cut-off. The pivot P, is located precisely as in the Hackworth gear, and the lever FP can be turned from "full gear forward", as shown in the figure, to "full gear backing", as shown by the line BP, intermediate positions give different cut-offs as with the Hackworth gear. Since FA is made equal to FP, the point A will always swing through P, no matter where F may be, and will coincide with P, when the engine is on dead center. The leads therefore will remain constant, as in the preceding case.

The Marshall gear is sometimes made with C at the right of A on a prolongation of the line EA. In this case if the same kind of valve is to be used, the eccentric E must move with the crank instead of 180° from it. The Marshall gear is frequently used on marine engines, the one eccentric being simpler than the two required by the Stephenson.

Joy Gear. Perhaps the most widely known, and certainly one of the best radial gears is the Joy, outlined in Fig. 46. It is frequently used on marine engines and on some English locomotives. No eccentrics are used, the valve motion being taken from C, a point on the connecting rod. H is a fixed pivot supported on the cylinder casting. The lever ED has a block pivoted at A, which slides back and forth in a curved slotted guide. The guide and the lever PF are fastened to the reverse shaft P, and by means of a reverse rod leading off from F, can be turned from full gear forward, as shown, to full gear backing when the pin F moves over to B. Motion is transmitted to the valve stem by means of the radius rod EG. The proportions are such that when the crank is on either dead point, the pivot of block A coincides

with P, so that the curved guide may then be set in any position without moving the valve; therefore the leads are constant. This gear gives a rapid motion to the valve when opening and closing and a more nearly constant compression than the Stephenson gear, and the cut-off can be made very nearly equal for all grades of the gear. Its many joints cause wear and its position near the crosshead, makes a careful inspection of the crosshead and piston exceedingly difficult while the engine is running.

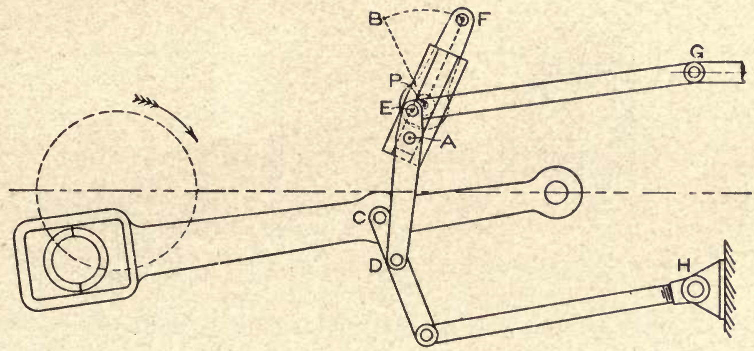

Fig. 46.

Walschaert Gear. This radial valve gear, although seldom seen in the United States, is the valve mechanism most commonly used on locomotives built on the continent of Europe. Like all other radial gears, it gives constant lead, and a distribution of steam very nearly alike for each end of the cylinder. In this respect it is superior to the Stephenson link, and gives without doubt better economy, but its mechanical construction is complicated, and not well adapted to the American type of locomotive. Fig. 47 illustrates this type of gear. S is the center of the driver axle. The crank pin K has forged on its center end an arm KE, on which the pin E is fixed. This arm lies parallel to the plane of the driving wheels, and being fixed to the crank pin, turns with the wheel, allowing the connecting rod to pass between it and the driving wheels. In this manner the point E moves around S in a circle, and moves the rod EH back and forth just as if it were an eccentric. It is so made that ES is perpendicular to the crank KS, and therefore the action of the pin E is equivalent to an eccentric with no angular advance.

This arm reaching back from the outer end of the crank pin is one of the most objectionable features on the construction, and is sometimes replaced by the regular type of eccentric put on the shaft between the driving wheels.

The eccentric rod EH causes the box link HP to oscillate on fixed trunions P. This link has a groove curved to a radius equal to GD, the length of the radius rod. A block pivoted at G, on one end of the radius rod, is free to move up or down in this

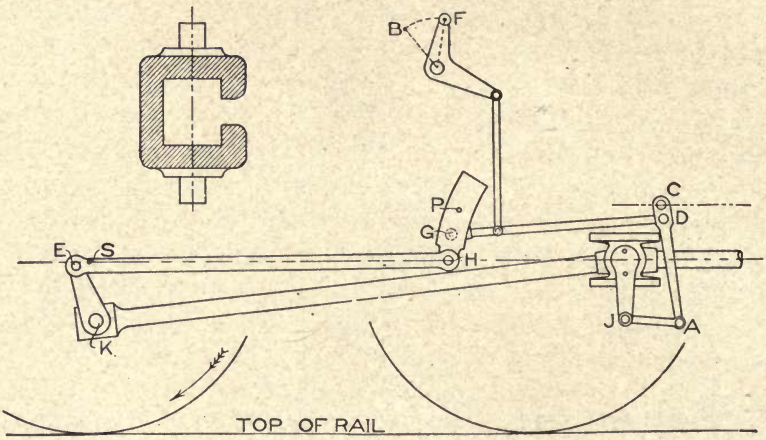

Fig 47.

groove. The valve derives its motion from C, a pivot on the floating lever CA. Point A receives motion from the crosshead; point D from the eccentric and the curved link; and a combination of these two imparts motion to C (which can slide only along the dotted line). A bell-crank lever pivoted above the link shifts the mechanism from "full gear forward" when F is moved to B, thus raising G above the link pivot or saddle pin.

ADJUSTABLE ECCENTRICS.

The position of an eccentric for a plain slide valve is 90° plus the angular advance ahead of the crank, in the direction in which the engine is to turn. Thus A, Fig. 48, is correctly placed, relative to the crank C, if the engine turns right-handed. For running in the opposite direction, the position of the eccentric is at D. Some engines are provided with a reversing mechanism which causes the eccentric to shift from A to D, either along the arc

ABD, or along the straight line AED. Such engines provide, not only for reversing, but for changing the cut-off as well If the eccentric moves on the arc to OB, the angular advance is

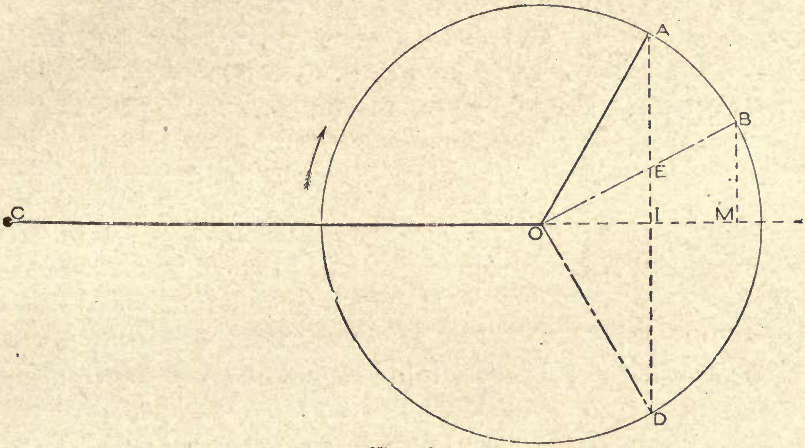

Fig. 48.

increased and all the events of the stroke are hastened as well as the cut-off, but the travel of the valve is not changed.

Zeuner's diagram, Fig. 49, is lettered to correspond with Fig. 48, and shows the effect of changing the angular advance from FOA to FOB. If OK represents the lap, the crank angle at cut-off will decrease from HOK to HOL, and the lead will increase very much, viz., from GF to HF. If the eccentric is shifted on the straight line to E (Fig. 48), a different valve motion will result. The angular advance is increased as before, so that all the events are hastened, but the eccentricity is now only OE instead of OB and the valve travel is consequently reduced. Zeuner's diagram for this case, Fig. 50, shows a decrease in crank angle at cut-off from IOM to ION, and no

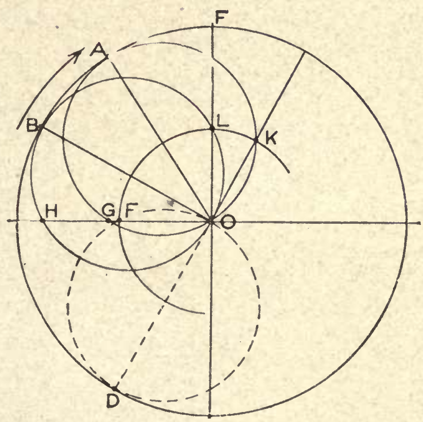

Fig. 49.

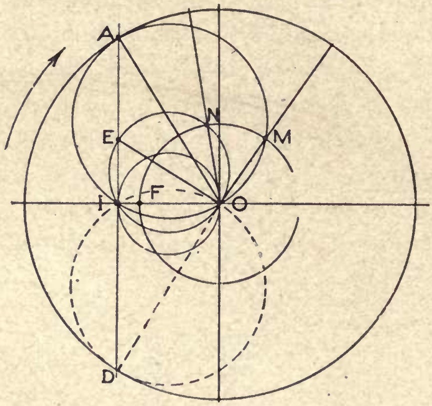

Fig. 50.

change in the lead IF. Let us consider the eccentric position OA, Fig. 48. In this position OI represents the displacement of the valve from mid-position when the engine is on center. If the eccentric moves to OB, the displacement will be OM, which is greater, showing an increase in lead equal to IM, but if the position is OE instead of OB, the displacement from mid-position will be OI as before. It is evident that the eccentric can move on the straight line from A to D, without changing the lead, while the decreased valve travel will result in an earlier cut-off. If the

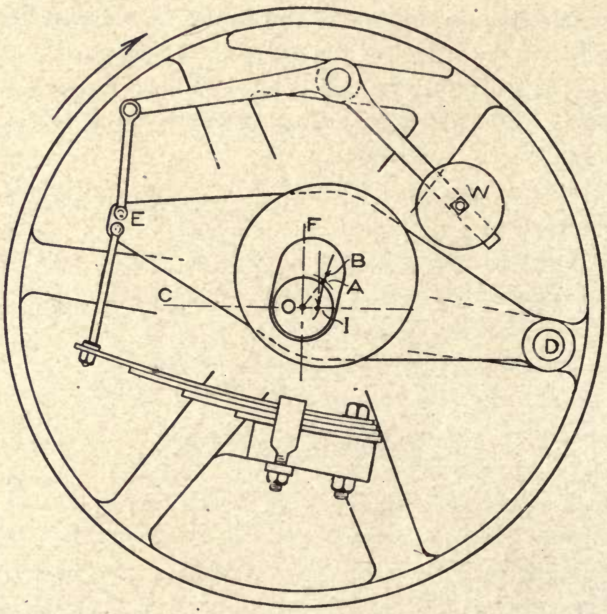

Fig. 51.

shifting eccentric is to be used for an automatic cut-off, as in the various types of fly-wheel governor, the curved path is not

desirable on account of excessive lead at short cut-off, but if it is to be used only as a means of reversing, it is preferable to the straight line.

All fly-wheel governors operate by shifting the eccentric, either to change the angular advance, the travel of the valve, or both. Fig. 51 illustrates the principle of a governor arranged to give decreasing lead, but as these mechanisms are described in the Steam Engine—Part I, under the head of governors, a further discussion will not be given here.

The device shown in Fig. 52 is often used for reversing engines of small launches. The eccentric E is loose on the shaft between a fixed collar G, and a hand wheel H. A stud projecting from the eccentric passes through a curved slot in the disc of the wheel, and can be clamped by a hand nut F. When running forward with the crank at C, the center of the eccentric is at A, and the nut clamped at F. To reverse, steam is shut off, and when the engine stops, the nut F is loosened, and then moved to B and clamped; or after F is loosened, the wheel, shaft, crank and propeller are turned over by hand until B strikes the stud at F, where it is clamped. The engine will then run astern.

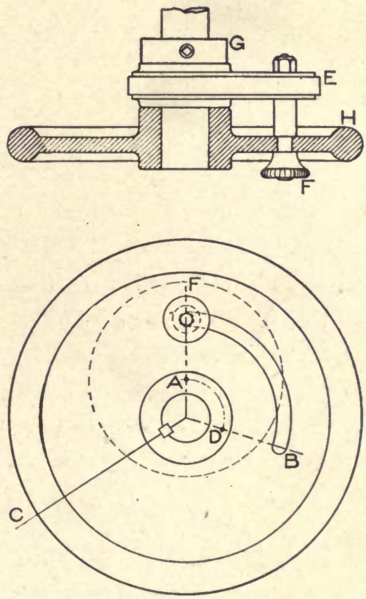

Fig. 52.

To study the application of the Zeuner diagram to this form of mechanism turn again to Fig. 51. If OA is the desired eccentricity for a normal position of the governor, the perpendicular distance of A from OF is made equal to the lap OI, plus the desired lead. Pivot D is then located equally distant from A and I. Zeuner's diagram for this gear, drawn to an enlarged scale, is shown in Fig. 53. The angular advance F'OB is laid back toward OC. OB is the

maximum eccentricity; OI, the lap or the desired least eccentricity. An arc, with proper radius, described through B and I shows the path of the eccentric. If the eccentric moves in to A, the crank angle at cut-off is decreased from COD to COE, and the lead decreased from FI to GI. A slight decrease in lead is

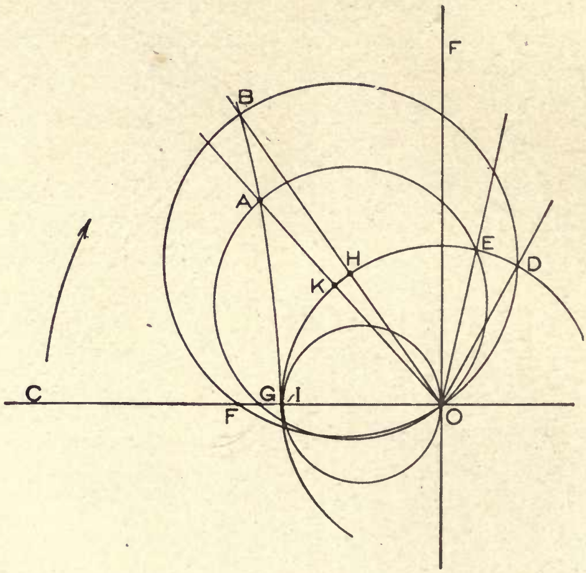

Fig. 53.

not objectionable, since the speed is not allowed to increase more than two or three per cent; and further, as the lead increases, compression decreases, so that one influence helps to counteract the other. The decrease in maximum port opening from BH to AK is unavoidable, but it is permissible, since it occurs only when the load decreases, and when less steam should be admitted to the cylinder.

DOUBLE VALVE GEARS.

It has been shown in the preceding discussion, that a plain slide valve under the control of a gear that gives a variable cut-off, such as a shifting eccentric or a link motion, will not give a satisfactory distribution of steam at short cut-off owing to execessive compression, variable lead, or early release. These difficulties are overcome in a measure by the use of the radial gear ; and also by the use of two valves instead of one. The main

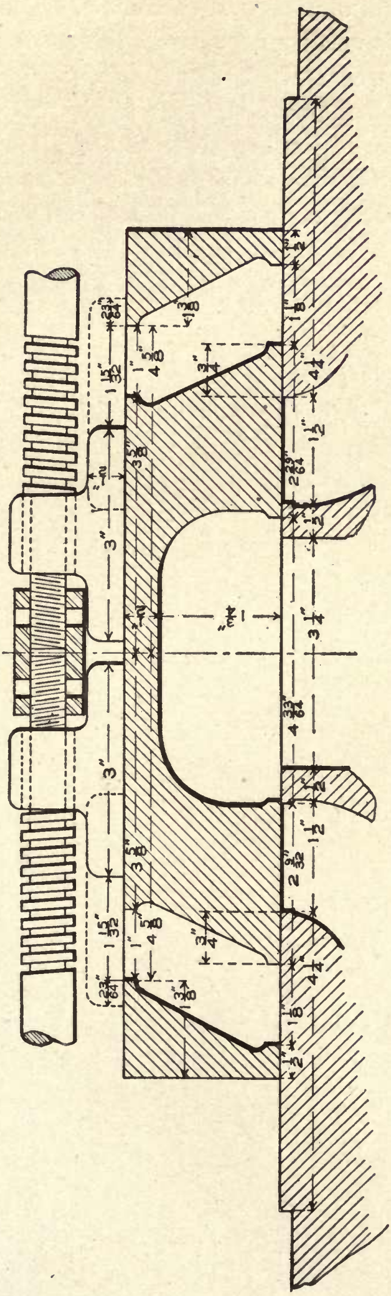

Fig. 54.

valve controls admission, release, and compression; the other, called the cut-off valve, regulates the cut-off only, which may be changed without in any way affecting the other events of the stroke. This cut-off valve may be placed in a separate valve chest, or it may be placed on the back of the main valve.

Meyer Valve. The most common form of double valve gear is the Meyer Valve, Fig. 54. The cut-off valve is made in two parts and works on the back of the main valve. The two parts are connected to a valve spindle with a right- and left-hand thread, so that their positions may be altered by rotating the valve spindle.

A swivel joint is usually fitted in the valve-spindle between the steam chest and the head of the valve rod, and the valve spindle prolonged into a tail rod which projects through a stuffing box on the head of the steam chest. See Fig. 55. The end of this tail rod is square in section and reciprocates through a small hand wheel by means of which it can be rotated while the engine is running, whatever the position of the valve may be.

Each valve is under the control of a separate eccentric. The eccentric moving the main valve is usually fixed, while the cut-off valve eccentric may be under the control of a governor. Since a slight compression is desired, the main valve is set to give late cut-off. This will give late release and late compression, and allow a wide range of cut-off for the cut-off valve. With this gear, lead, release, and compression are entirely independent of the ratio of expansion, and the cut-off is much sharper, because the cut-off valve, when closing the ports, is always moving in a direction

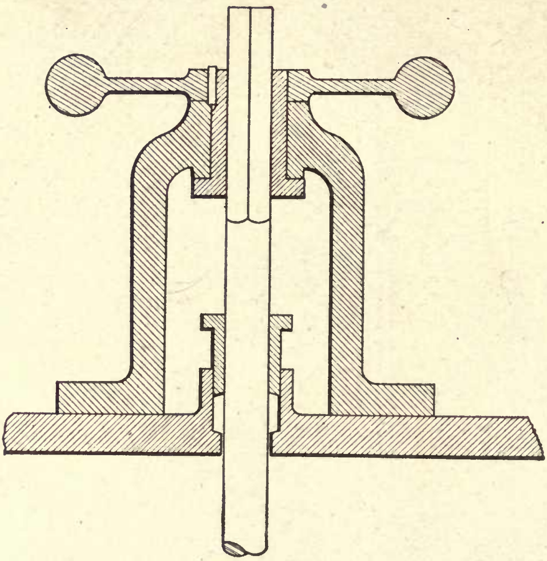

Fig. 55.

opposite to that of the main valve. The valve may be designed by means of Zeuner's diagrams.

Design of a Meyer Valve. Let us design a Meyer Valve having an eccentricity of 2 inches. Let the eccentricity of the cut-off valve be $2\frac{1}{4}$ inches and the relative travel of the cut-off valve in relation to the main valve be 3 inches. This will make the relative motion of the cut-off valve equivalent to the travel of a plain slide valve with an eccentricity of $1\frac{1}{2}$ inches. Let the outside lap on the main valve be $\frac{3}{4}$ inch, the lead $\frac{1}{32}$ inch, the compression 95 per cent of the stroke, and let the ratio of the length of the crank to connecting rod be six.

In Fig. 56 draw XOY equal to 4 inches, the main valve travel. Lay off YD = 95 per cent of 4 = 3.8 inches, and with a radius of 12 inches, and the center on YX produced draw the arc DH.K. H.K.O is the crank position at compression. C.K.O, the crank position at cut-off, is found in a similar manner. Lay off OI equal to the lap plus the lead, and draw the valve circle for the main valve through I and O with a diameter equal to its eccentricity of 2 inches. To do this take a radius equal to 1 inch, and draw arcs from I and O as centers that shall intersect at B. B is

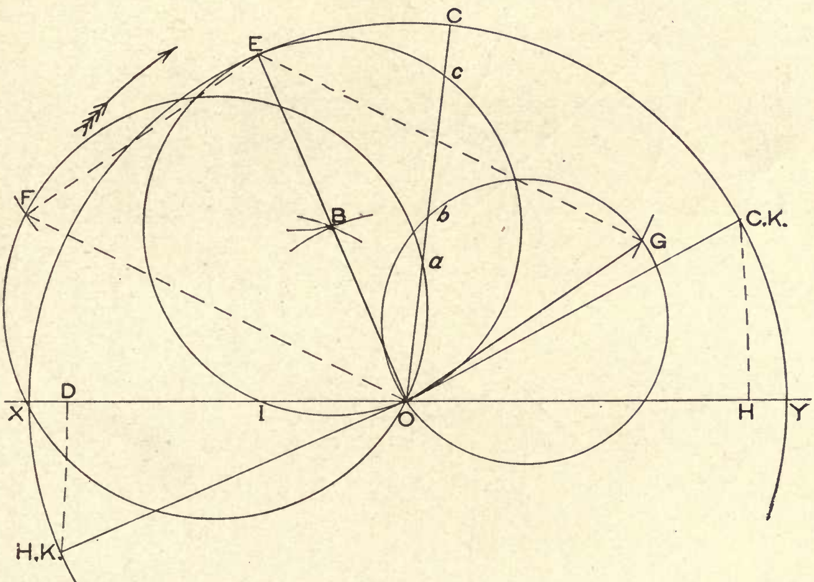

Fig. 56.

the center of the valve circle and OBE is the eccentricity, 2 inches. With E as a center, and with a radius equal to half the relative travel of the cut-off valve (in this case $1\frac{1}{2}$ inches), draw an arc. With O as a center and with a radius equal to $2\frac{1}{4}$ inches, the eccentricity of the cut-off valve, draw another arc intersecting the first one at F. On OF as a diameter construct a valve circle. This valve will represent the absolute motion of the cut-off valve, independent of the motion of the main valve. This circle then will show the displacements of the cut-off valve from the center of the steam chest. With E as a center and with a radius equal to FO draw an arc, and with O as a center and with a radius equal

to EF draw another arc intersecting the first at G. On OG as a diameter construct a valve circle. This circle will then represent a travel of the cut-off valve moving on the main valve. That is, it will represent the displacements of the cut-off valve from the center of the main valve. This circle is not, properly speaking,

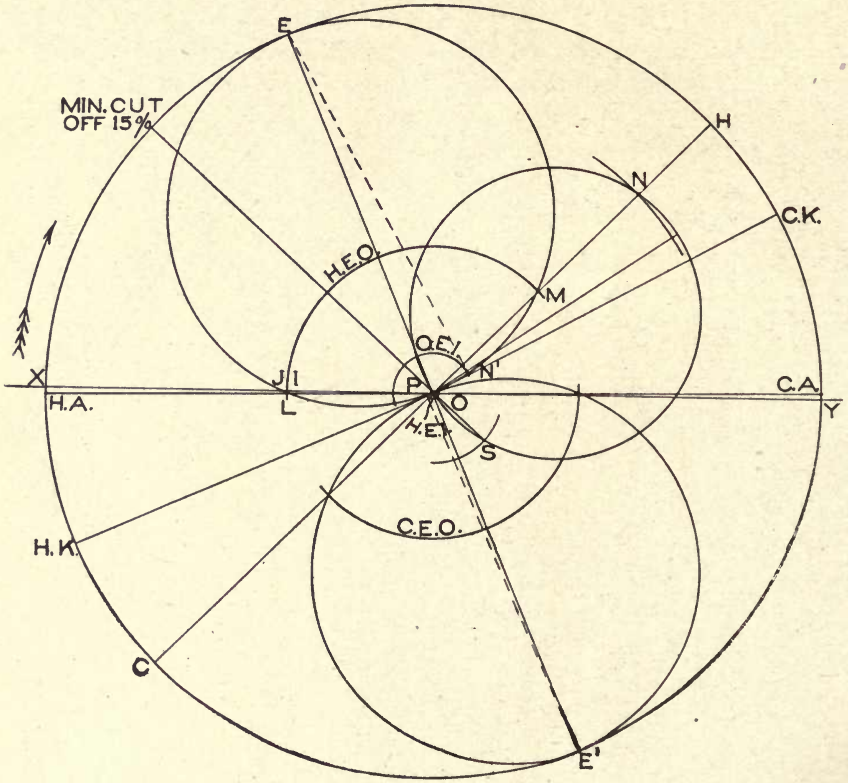

Fig. 57.

a valve circle and OG is not an eccentricity, but simply represents the relative motion of the two valves. This can be proved by analytical geometry, but an inspection of the figure shows that this must be true.

Draw the crank line OC at any position, cutting the valve circles at *a* and *b* and *c*. O*a* represents the absolute displacement of the cut-off valve, that is, from the center of the steam chest and O*c* represents the displacement of the main valve. The

relative displacement of the cut-off valve, that is, from the center of the main valve, will be the difference between Oc and Oa, since both valves are moving in the same direction. By careful measurement it will be found that $Ob = Oc - Oa$, and any arc as Ob on the auxiliary circle ObG will correctly represent the displacement of the cut-off valve from the center of the main valve at the corresponding crank angle.

Fig. 57 shows the crank angle at head-end compression H.K., and at crank-end compression C.K., the main valve circle, and the auxiliary circle which are transferred from Fig. 56. The construction lines and all lines not essential to the figure are omitted to avoid confusion.

Lay off on Fig. 57, OI equal to the outside lap $\frac{3}{4}$ inch and draw the head-end lap circle H.E.O. It will intersect the valve circle for the main valve at L and M. Through L draw the crank position at admission (head-end) H.A. and the crank position at cut-off through M. This gives the greatest possible cut-off. The cut-off valve may be set to give a much earlier cut-off than this, but of course a later setting would be of no avail for the port would be closed by the main valve at this angle. The crank line OMH cuts the auxiliary circle at N, so that ON ($1\frac{15}{32}$ inches) is the clearance of the cut-off valve. That is, the edge of the cut-off valve must be set $1\frac{15}{32}$ inches from the edge of the main valve port in order to cut-off at this crank angle. The full lines of Fig. 54 show the cut-off valve placed in this position.

The intersection of H.K.O with the lower valve circle, gives the inside lap at the head end of the cylinder. This line comes so nearly tangent to the valve circle that the intersection can be determined only by dropping a perpendicular to H.K.O. from E^1. This cuts the circle at P and $OP = \frac{1}{32}$ inch equals the head-end inside lap, and H.E.I. represents the corresponding lap circle.

The crank-end angle at compression is C.K. which cuts the upper valve circle at N′, giving an inside lap for the crank end of $ON' = \frac{13}{64}$ inch. To make this intersection more apparent the perpendicular can be drawn from E as previously explained.

Suppose that it was required that the minimum cut-off should be 15 per cent. Find the crank position at 15 per cent of the stroke in the same manner as the crank position was found at compression,

Produce this line through O until it cuts the auxiliary circle at S. Then OS $= \frac{23}{64}$ inch = the required lap for the cut-off valve in order to cut-off at 15 per cent of the stroke. The dotted lines in Fig. 54 show the cut-off valve drawn in this position.

For a valve of this sort, the cylinder port should be $1\frac{1}{2}$ inches wide and the valve port 1 inch wide. Fig. 54 shows this valve laid out to scale, but as this process is in all respects similar to that described for laying out a plain slide valve, it will not be described in detail.

DROP CUT-OFF GEARS.

The ordinary slide valve controls eight different events of the stroke, that is, admission, cut-off, release, and compression for both ends of the cylinder. A change in the setting of a plain slide valve that affects any one event on the crank end, let us say, will also change to a greater or less degree every other event of the stroke, on the head end as well as on the crank end; so that in setting a slide valve the desired position for one event must usually be sacrificed in order to make the others less objectionable.

In order to provide a better distribution of steam than is possible with a single valve, some engines have four valves, two at each end of the cylinder. In horizontal engines, two are placed above the center line of cylinder and two below. The upper are for admission and cut-off, the lower for release and compression. Since each valve controls but two events, a very satisfactory adjustment can be made and the extra complication and cost for large engines are more than overbalanced by the advantages gained, viz.: A very much better distribution of steam, short steam passages and small clearances, separate ports for the admission of hot steam and the exhaust of the same steam after expansion when its temperature has fallen, and finally the possibility of opening and closing the ports very rapidly, thus preventing wire-drawing. The small clearances, short ports and separate admission and exhaust materially reduce the cylinder condensation, and thus effect a large saving in the steam consumption.

When four valves are used for high speeds, the motions of all must be positive, that is, they must be connected directly to some mechanism that either pushes or pulls them through their entire motion, but for speeds up to 100 revolutions or so a disen-

gaging mechanism may be used, and the valves may shut of themselves, either by virtue of their weight or by means of springs or dashpots. The valve is usually opened by means of links or rods, moved by an eccentric, and at the proper point of cut-off the rod is disengaged from the valve which drops shut, hence the term "drop cut-off" gears.

Reynolds-Corliss Gear. The most widely known drop cut-off gear is the Reynolds-Corliss, shown in Figs. 58 and 59; it is often referred to as the *Reynolds hook-releasing* gear. An eccentric on the main shaft gives an oscillating motion to a circular disc

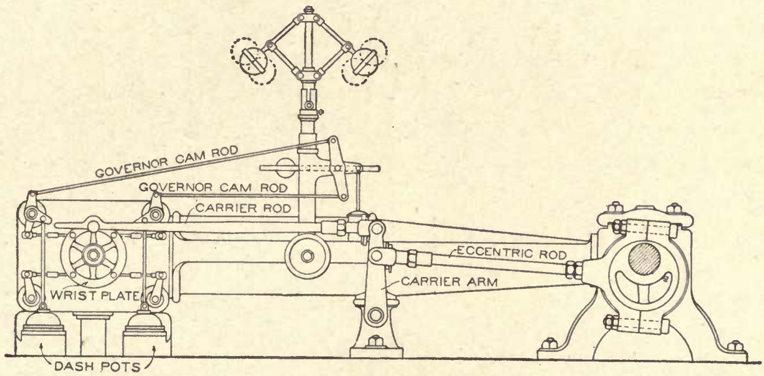

Fig. 58.

called the wrist plate, pivoted at the center of the cylinder. It transmits motion to each of the four valves through adjustable links known as *steam rods* or *exhaust rods*, according to whether they move the admission or exhaust valves.

The valves which are shown in section in Fig. 60 oscillate on cylindrical seats, and the position of the *rods* is so determined that they give a rapid motion to the valve when opening or closing, and hold it nearly stationary when either opened or closed.

The Reynolds hook is shown in detail in Fig. 59. The *steam arm* is keyed to the valve spindle which passes loosely through a bracket on which the *bell-crank* lever turns, and the spindle is packed to make a steam-tight joint where it enters the cylinder. Motion of the *steam rod* toward the right will turn the *bell-crank* lever and raise the *hook stud.* The *hook* (from which the gear derives its name) pivoted on this stud, has at one end a hard-

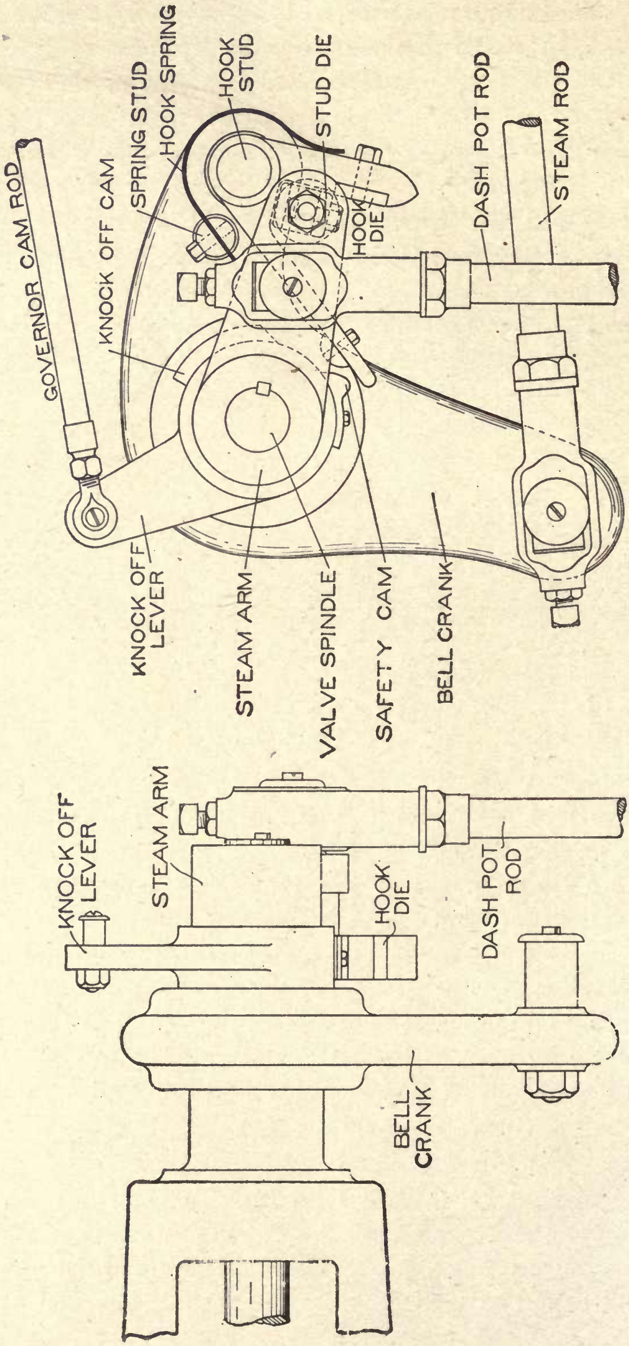
GOVERNOR CAM ROD
KNOCK OFF CAM
SPRING STUD
HOOK SPRING
HOOK STUD
STUD DIE
HOOK DIE
DASH POT ROD
STEAM ROD
KNOCK OFF LEVER
STEAM ARM
VALVE SPINDLE
SAFETY CAM
BELL CRANK
KNOCK OFF LEVER
STEAM ARM
HOOK DIE
DASH POT ROD
BELL CRANK

Fig. 59.

ened steel die with sharp, square edges, and at the other end, a small steel block with a rounded face. As the hook rises, the *hook die* engages the *stud die* which is fastened to the *steam arm*, and one end of the steam arm is thus raised. This turns the valve in its seat and admits steam. As the *hook* continues to rise, its stud moves in an arc above the valve spindle, and the round-faced block at its left-hand end strikes the *knock-off cam* which causes the *hook* to turn about its stud and disengage the *hook die* from the *stud die.* In raising the *steam arm*, the *dash-pot rod* also is raised and a partial vacuum is created in the dash-pot. As soon, therefore, as the dies become disengaged, the *dashpot rod* quickly drops under the force of this vacuum, thus turning the steam arm and closing the valve. The striking of the left-hand end of the hook against the knock-off cam determines the point of cut-off, by releasing the valve at that instant.

This cam is a part of the *knock-off lever* to which the *governor cam rod* is fastened. Any action of the governor which would cause the *cam rod* to move toward the right would cause this *knock-off lever* to turn on its axis, the steam arm, and consequently lower the position of the *knock-off cam.* This would cause an earlier contact between the cam and end of hook, and consequently an earlier cut-off. By lengthening or shortening the *governor cam rod*, the point of cut-off can be adjusted to suit the engine load without changing the speed.

There is a limit to this adjustment, for it can be shown that a Corliss gear operated by a single eccentric cannot be arranged to cut-off later than half stroke. Suppose the eccentric is set just 90° ahead of the crank. Then it will reach its extreme position just as the piston gets to half stroke. If by that time the *hook* which was rising and opening the admission valve, has not yet struck the *knock-off cam*, it cannot strike it at all, for any further motion will cause the hook to descend to its original position, that is its position at the beginning of the stroke; the hook will not disengage from the steam arm *stud* at all and the bell crank will return, closing the valve in the same manner in which it opened it. Cut-off will then take place near the end of the stroke, but it will not be the sharp cut-off produced by the sudden *drop* when the dies are disengaged.

If the eccentric were set less than 90° ahead of the crank, the cut-off could be arranged to occur later than half stroke, but this is decidedly impracticable, for with such a position of the eccentric the action of the valves at release and compression is spoiled. When it is necessary to cut-off later than half-stroke, as sometimes happens on low-pressure cylinders of compound engines, it may be arranged for by means of two eccentrics, one set more than 90° ahead of the crank to operate the exhaust valves, and one less than 90° ahead to operate the admission valves.

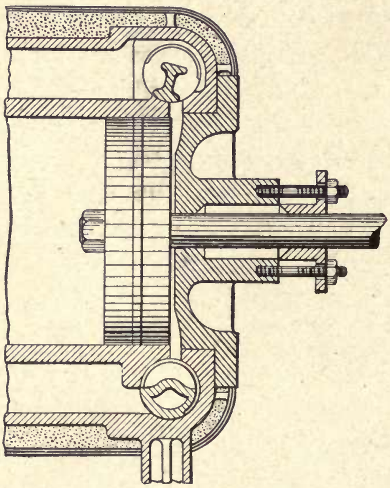

Fig. 60.

The safety cam shown in Fig. 59 is an important part of a Corliss gear. If for any reason the engine governor should fail to act, due, for instance, to the breaking of its driving belt, the governor would drop to its lowest position, supply more steam to the engine and allow it to run away. The *safety cam* prevents this by moving so far to the right that it strikes the hook when it descends to pick up the steam arm. The hook is consequently turned toward the right and then lifted without engaging the stud die; the valve consequently remains closed and the engine stops.

Brown Releasing Gear. In addition to the *Reynolds hook*, several other devices are in use for opening and releasing Corliss admission valves. Among them the Brown releasing gear shown in Fig. 61 may be noted. The steam rod and dashpot rod are arranged much the same as in the Reynolds gear. The governor cam rod operates a plate cam having a curved slot so shaped that it takes the place of both the knock-off and the safety cam of Fig. 59. The steam arm is keyed to the valve spindle and carries at its lower end a steel die which is free to slip up and down a small amount. The part of this gear corresponding to the Reynolds bell crank becomes a straight rocker pivoted at its middle;

and the part corresponding to the Reynolds hook has at one end a die which engages the die of the steam arm, and at its other end a roller running in the curved cam slot. This hook is really a bell-crank lever with arms that are not in the same plane. The bearing on which it turns is carried on the lower end of the rocker, and therefore is equivalent to a movable pivot similar to the hook stud of the Reynolds gear.

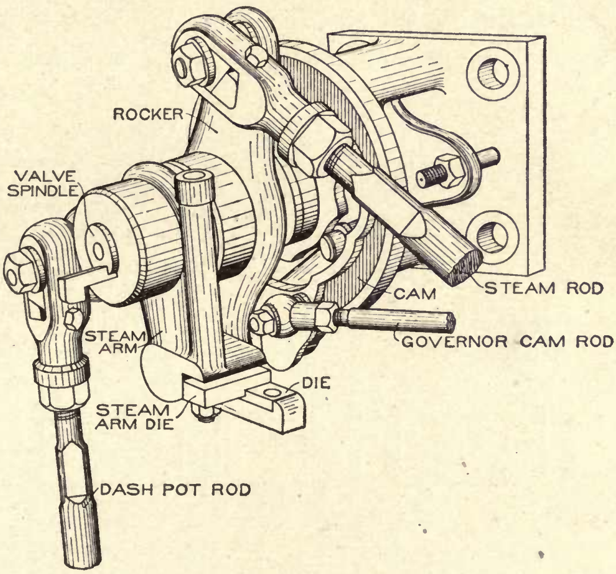

Fig. 61.

In the position shown the dies are engaged. Motion of the steam rod toward the right will move the lower end of the rocker toward the left, and consequently turn the valve spindle in a right-handed direction. This will open the valve and at the same time raise the dashpot rod. Meanwhile, the roller is moving toward the left in a circular part of the cam slot, the center of which is at the center of the valve spindle. This causes the steam arm and the bell-crank lever, which has the roller at one end, to move in such a way that there is no relative motion between them. As soon, however, as the roller comes to the point where it is forced to move out of this circular path and move farther from the valve spindle, both arms of the bell-crank lever are turned downward,

the dies become disengaged, and the dashpot closes the valve. The slight up-and-down motion of the steam-arm die allows it to rise while the hook die passes underneath when returning to re-engage for the next stroke. The makers claim that this gear permits a much higher speed than is possible with other Corliss gears.

Greene Gear. Another well-known drop cut-off gear is the Greene, shown in Fig. 62. The valves are of the gridiron type, sliding on horizontal seats, the admission valves parallel to, and the exhaust valves at right angles to the axis of the cylinder and just below it. AA are rock shafts turning in fixed bearings.

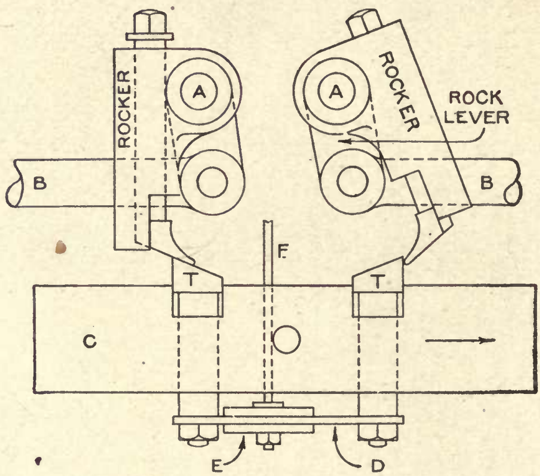

Fig. 62.

BB are the admission valve stems. C is a slide bar, receiving a reciprocating motion from an eccentric. TT are tappets connected to the slide bar. They move to and fro with the slide bar and can also move independently up and down. They are made fast at their lower end to the gauge plate D which slides through the guide E. The guide E is made fast to the governor rod F and through this means can be raised or lowered, thus regulating the height of the tappets.

As the slide bar moves toward the right, the right-hand tappet is brought into contact with the toe of the rocker, causing it to turn on its bearings and move the rock lever and the valve stem B toward the right, thus opening the admission valve. Since

the tappet moves in a horizontal direction while the toe of the rocker moves in an arc, it will be seen at once that they will soon become disengaged and release the valve which is at once closed by a dashpot not shown in the figure. If the governor raises the tappets, cut-off will be later. A nut at the bottom of the governor rod allows a proper adjustment of the guide and gauge plate. As the slide plate C moves toward the right, the left-hand tappet comes in contact with the heel of the left-hand rocker, both being beveled, it rises in its socket allowing the tappet to pass under. It then falls by its own weight and is ready to engage the tappet on its

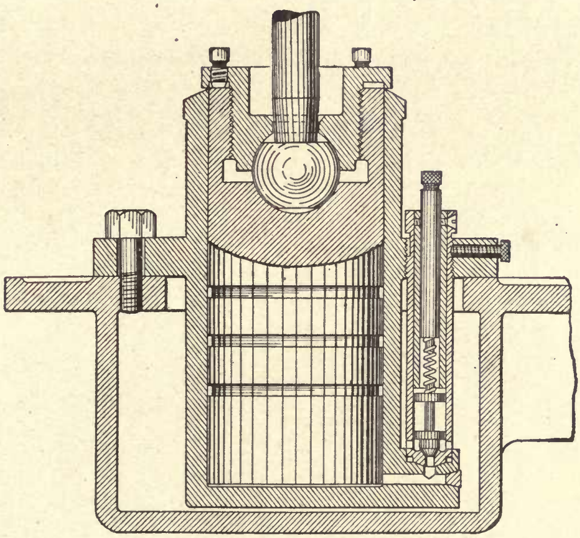

Fig. 63.

return and open the valve. In this gear the disengagement of the valve throws no load whatever on the governor which is a distinct advantage over the Corliss gear. The action of the exhaust valves is not shown in the cut.

The Sulzer Gear is a drop cut-off widely used in Europe. The valves are of the poppet type, lifting straight from conical seats, so that there is no friction. They are usually placed vertically above and below the cylinder axis and are operated by eccentrics from a shaft geared to the main shaft. The admission valves are lifted from their seats by suitable levers, then released by a

device equivalent in action to the Reynolds hook and are quickly closed by the action of springs.

The exhaust valves of all drop cut-off gears are comparatively simple in their operation and both in opening and closing they are moved by the direct action of the exhaust rods.

A common form of vacuum dashpot for closing admission valves is shown in Fig. 63. The rod coming down from the steam arm makes a ball-and-socket joint with the dashpot piston. The dashpot is often let down into the engine frame as shown. When lifted, the piston produces a partial vacuum underneath it so that it tends to drop quickly as soon as the valve gear is released. On some of the largest modern engines where the valves are very heavy, steam-loaded dashpots are used; that is, the dashpot piston has steam pressure on one side, and an air cushion on the other prevents it from striking the bottom of the dashpot.

Corliss Valve Setting. The setting of a Corliss valve gear is a much longer process than the setting of a plain slide valve, but is nevertheless a comparatively simple matter, for the various adjustments are nearly all independent of one another. In gears like that shown in Fig. 58 the length of both the eccentric rod and carrier rod are unusually adjustable, and the former should be of such length that the carrier arm swings equal distances on each side of a vertical line through its pivot, and the carrier rod should be adjusted until the wrist plate oscillates symmetrically about a vertical line through its pivot Nearly all Corliss engines have one mark on the wrist plate hub and three on the wrist plate stand, as shown in Fig. 64, and the wrist plate should swing so that A, the mark it carries, moves from C to D, but not beyond either one. When A is in line with B, the wrist plate is in mid-position. The valves are then not in their exact mid-position, but it is customary to regard them as being in mid-position, and to speak of the lap as the amount the valve covers the port when the wrist plate is in mid-position.

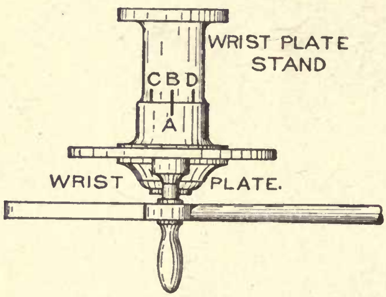

Fig 64.

To set the valves, remove the bonnets or covers of the valve chambers on the side opposite the gear. The ends of the valves are circular, but inside their cross-section is as shown in Fig. 65. On the end, in line with the finished edge of the valve, and on the seat in line with the edge of the steam port, are marks as shown in Fig. 65. When these marks coincide, the valve is either just opening or just closing, and when in any other position, the amount of opening or the amount by which the port is closed is shown directly by the distance between the marks. Block the wrist plate in mid-position, hook up the admission valves and adjust the length of the steam rods by means of the right and left threads provided for the purpose, until the ports are covered by the amount of lap indicated in the following table opposite the given size of engine.

Dia. of Cyl. in inches.	Steam Lap in inches.	Exhaust Clearance in inches.
12	$\frac{1}{4}$	$\frac{1}{32}$
14 to 16	$\frac{5}{16}$	$\frac{1}{32}$
18 to 22	$\frac{3}{8}$	$\frac{1}{16}$
24 to 28	$\frac{7}{16}$	$\frac{3}{32}$
30 to 36	$\frac{1}{2}$	$\frac{1}{8}$
36 to 42	$\frac{5}{8}$	$\frac{5}{32}$

Next adjust the exhaust rods until the exhaust ports are open an amount equal to the clearance given in the above table. Set the engine on its head-end dead point, hook the carrier rod onto the wrist plate and in the direction in which the engine is to run, turn the eccentric enough to open the head-end admission valve by a proper amount of lead; then the eccentric will be 90° plus the angular advance ahead of the crank. The proper amount of lead will depend upon both the design of the gear and the speed at which the engine is to run; and may vary from $\frac{1}{32}''$ for small engines to as much as $\frac{5}{32}''$ or $\frac{3}{16}''$ for large and higher-

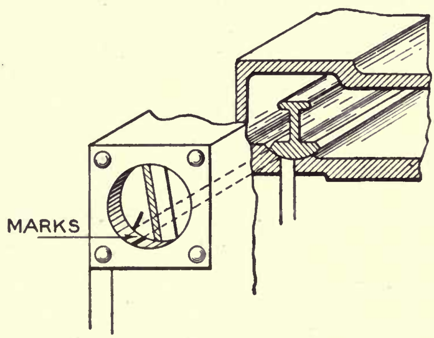

Fig. 65.

speed engines. When the proper amount of lead has been obtained, fasten the eccentric on the shaft by means of the set screw and make sure by trial that the wrist plate moves to its extremes of travel. The dashpot rods must be adjusted so that when the dashpot piston is at its lowest position, the hooks (see Fig. 59) descend just far enough for the hook dies to snap over the stud dies with about $\frac{1}{32}''$ to $\frac{1}{16}''$ to spare, depending on the size of the gear.

To adjust and equalize the cut-off, lift the governor to about the position that it will occupy when running at normal speed, and put a block under the collar to hold it in this position. First set the double lever at the right of the governor cam rods so that it makes approximately equal angles with each rod, and then turn the engine over by hand until the piston has moved to the desired point of cut-off. Adjust the proper cam rod until the knock-off cam strikes the hook and allows the valve to close, then turn the engine to the point of cut-off on the other stroke and adjust the other cam rod in a similar manner. Now set the governor in the lowest position to which it could fall if there were no load on the engine, and set the safety cams so that in this position the hook cannot open the valve. A latch is provided on which the governor can be supported slightly above its lowest position so that the valves can be opened by the hook when starting the engine. As soon as the engine speeds up this latch must be moved aside, so that if the governor fails to act, it can drop to its lowest point, otherwise this latch would hold it just high enough so that the safety cams could not act.

When Corliss gears are set as here described, the position of the eccentric may not be quite right, due to an incorrect estimate of the amount of lead required. The error is likely to produce faulty release and compression as well as poor admission, but it cannot be very serious, and the engine will turn over with its own steam, so that indicator diagrams may be taken. The final adjustments can then be determined from an examination of the diagrams.

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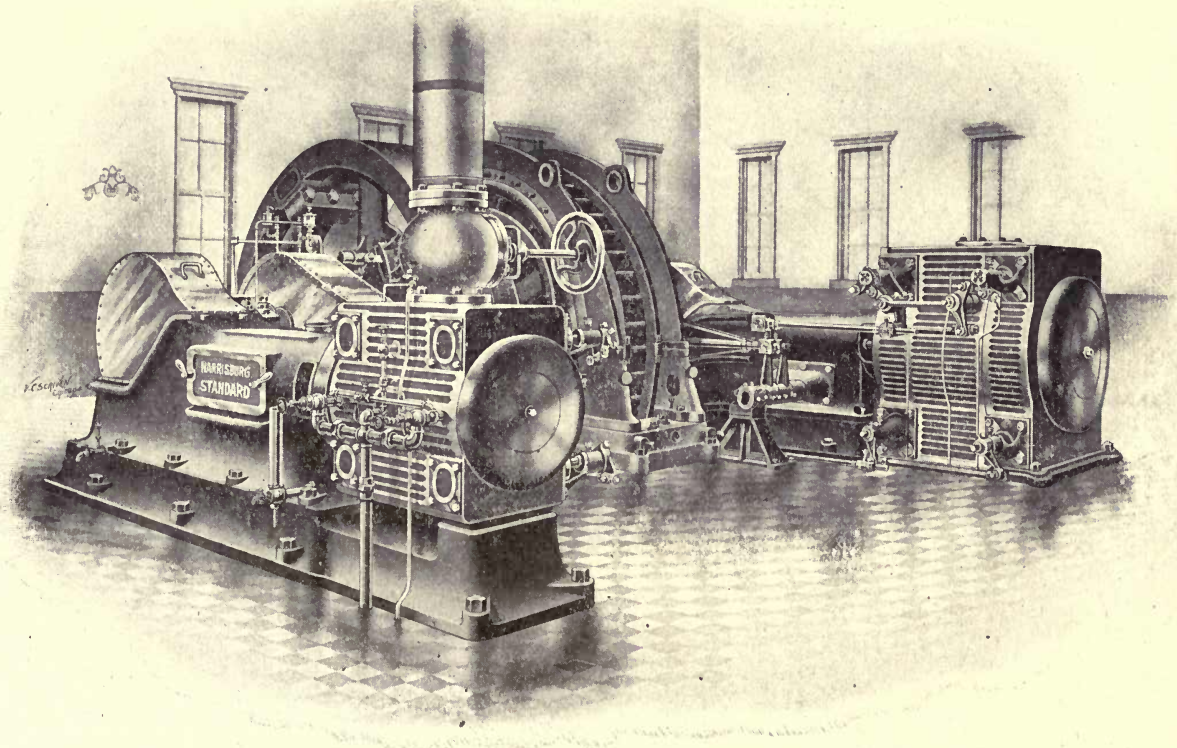


GENERAL CONSTRUCTION FLEMING CROSS COMPOUND FOUR-VALVE ENGINE DIRECT-CONNECTED STYLE, GENERAL VIEW.
Harrisburg Foundry and Machine Works.

STEAM ENGINE INDICATORS

A most important question concerning a steam engine is, "What is its horse-power?" or "How much work will it do in a given time?"

Work is defined as *pressure*, *force*, or *resistance* multiplied by the *distance* through which it acts.

Power is work done in a specified time.

In the steam engine, steam is the agent by means of which heat is transformed into mechanical work. It is the heat in the steam that does the work, not the steam itself.

Work is obtained from the heat in steam by confining it in a closed cylinder which is fitted with a piston and a piston-rod. Steam is admitted at one side of the piston while the other is open to the atmosphere or in communication with a condenser. The pressure of steam, usually 75 to 150 pounds per square inch, forces the piston to the other end of the cylinder, driving out the low-pressure steam in front of it. When it arrives at the other end, steam is admitted to that end and the piston is driven back.

The piston moves because the pressure on one side is greater than that on the other.

In order to move the piston, work must be performed. The amount of work is easily found, since work equals total pressure multiplied by the distance through which the piston moves.

Suppose a piston is 2 square feet in area and steam at a pressure of 64.7 pounds per square inch acts on it during the entire stroke of 4 feet; the other side of the piston being in communication with the atmosphere. The total pressure is then $2 \times 144 \times 64.7 = 18{,}633.6$ pounds. If this pressure acts through 4 feet it is evident from the definition that the work done per stroke will be,

$$18{,}633.6 \times 4 = 74{,}534.4 \text{ foot-pounds.}$$

Another method is as follows: The pressure on the above piston is $64.7 \times 144 = 9316.8$ pounds per square foot. The volume swept by the piston during one stroke is $2 \times 4 = 8$ cubic feet. If we multiply the pressure per square foot by the volume or 9316.8 by 8, we get 74534.4 foot pounds, the same result as before. Thus we see that work equals **unit pressure** multiplied by **volume.**

Let P = pressure on the piston in pounds per square foot.
p = pressure on the piston in pounds per square inch.
A = area of piston in square inches.
L = length of stroke in feet.
V = volume swept by piston in one stroke in cubic feet.
W = work done in foot-pounds.

Then from the above example,

Work = unit pressure multiplied by volume.

or, $W = P \times V$

It is evident that $P = 144\,p$, and $V = \frac{A}{144} \times L$.

Then we have these expressions for work,

$$W = P \times V = 144\,p \times V = 144\,p \times \frac{A}{144} \times L = p\,L\,A.$$

Suppose steam is admitted to the cylinder during the whole stroke, as in the above example, that is, one end of the cylinder is in communication with the boiler. The other end is open to the atmosphere. If we draw two lines at right angles to each other, as O Y and O X in Fig. 1, the volume of steam for any position may be represented by some distance measured on the line O X. Similarly the pressure of the steam at any position of the piston may be represented by the length of a vertical line parallel to the line O Y.

In the above example, the area of the piston was 2 square feet, the length of stroke 4 feet and the pressure by gage 50 pounds. Then we let O A = the atmospheric pressure = 14.7 pounds. At the beginning of the stroke the pressure (absolute) is $14.7 + 50 = 64.7$ pounds, represented by the distance O B, or A B = 50 pounds pressure. When the piston has passed through $\frac{1}{4}$ of the stroke it is represented as the point 1, or B 1 is the volume swept through when the piston has completed $\frac{1}{4}$ of the

stroke. At this point the pressure is also 64.7 pounds as represented at 1′. Similarly, when the piston is at 2, 3, and 4 the corresponding pressure is 2′, 3′, 4′. Since the pressure is constant the line B D is parallel to O X. We see from the above that 50 pounds is the net pressure acting on the piston during the stroke, and is represented by A B and lines parallel to it. The volumes are represented by the horizontal line A C. Then since $W = P \times V$ it also equals O B $\times$ O X which is evidently the area of the rectangle O B D X. The area of the rectangle O B D X is proportional to the work done by the steam.

In Fig. 1, one inch on the line O Y = 40 pounds, then O B is 1.6175 inches long since it represents 64.7 pounds. Similarly O A must be .3675 inch since it represents 14.7 pounds. The line A C is 2 inches long; then referring to the preceding example, one inch in length $= \frac{8}{2} = 4$ cubic feet.

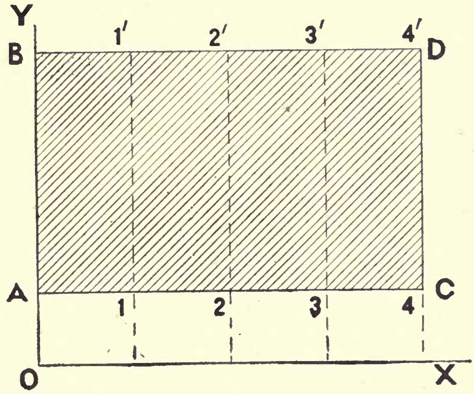


Fig. 1.

Since the rectangle O B D X is 1.6175 by 2 inches, the area is 3.235 square inches. But one inch in height equals 40 pounds pressure and one inch in length equals 2 cubic feet. Then $p\ V = 40 \times 3.235 \times 4 = 517.6$ foot-pounds and,

$$W = 144\ p\ V = 517.6 \times 144 = 74{,}534.4 \text{ foot-pounds.}$$

In the above cylinder the pressure acting on one side of the piston was 64.7 pounds per square inch. There was also a pressure of 14.7 pounds per square inch (the atmospheric pressure) acting in the opposite direction. Then the work done against the steam pressure is represented by the area O A C X and is equal to $144\ p\ V = 144 \times 14.7 \times 8 = 16934.4$ foot-pounds. Then since O B D X represents the total work done on one side of the piston and O A C X represents the work done *against* the piston the difference A B D C represents the **net work.** This net work is represented by the shaded area. Also if the amount of work done

on the piston is 74,534.4 foot-pounds and the work done against the piston is 16,934.4 foot-pounds, the net work is the difference, or 57,600 foot-pounds.

In this theoretical discussion the same result may be obtained by subtracting the atmospheric pressure or back pressure from the absolute initial pressure and using the difference as the value of p. This value of p is called the mean effective pressure.

Then $64.7 - 14.7 = 50$ and

$W = 144\,p\,V = 144 \times 50 \times 8 = 57{,}600$ foot-pounds.

The area is proportional to the work done whatever the shape may be; provided the line B D represents the relation between pressures and volumes on the steam side of the piston and the lower line A C represents the relation between pressures and volumes on the exhaust side. If the engine is of the condensing type the line A C will be nearer O X, which is the line representing *absolute vacuum.*

Whatever the shape of the diagram, the area is equal to the area of a rectangle of the same length and a height equal to the mean height, or mean ordinate as it is called. The mean ordinate represents the mean or average net pressure on the steam side of the piston. Then we can follow these rules in finding the work of the steam from the diagram.

Multiply the area in square inches by the scale of pressures, by the scale of volumes and by 144, *or;*

Multiply the length of the mean ordinate by the scale of pressures, by the length of stroke, and this product by the area of the piston in square inches.

Example: The area of a diagram A B D C like that of Fig. 1 is 6.3 square inches and its length is 3 inches. The scale of pressure is 30 pounds per inch and the scale of volumes is 1.99985 cubic feet to the inch. If the piston is 20 inches in diameter and the length of stroke $2\frac{3}{4}$ feet, what is the work done per stroke?

Solution:

W = area of diagram × scale of pressures × scale of volumes × 144.

$= 6.3 \times 30 \times 1.99985 \times 144 = 54{,}428$ foot-pounds.

W = mean ordinate × scale of pressures × area of piston × length of stroke.

$= 2.1 \times 30 \times 314.159 \times 2\frac{3}{4} = 54{,}428$ foot-pounds.

Thus we see that we get the same result by both rules. The latter is the more common method because the mean ordinate is easily found and the scale of volumes seldom considered.

In our consideration of Fig. 1, steam was admitted to the cylinder during the *entire stroke*. In modern engines this method is rarely used; instead, steam is admitted during *part* of the stroke then the communication to the boiler is cut off, and the steam in the cylinder allowed to **expand**, as the piston moves forward, until it fills the entire volume of the cylinder. This is represented graphically in Fig. 2.

Steam is admitted to the cylinder until the piston reaches the point 2 which represents one-half the volume of the cylinder. Then the cylinder is half full of steam, that is, it contains $\frac{8}{2} = 4$ cubic feet. The four cubic feet of steam expand until they fill the cylinder. Since there is the same weight of steam present at every point in the stroke and the volume continued to increase, the pressure must diminish.

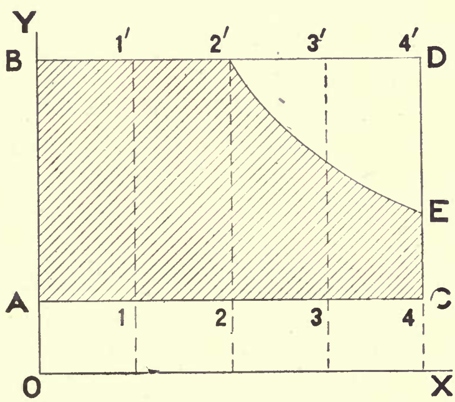


Fig. 2.

This is shown in Fig. 2. The line B 2′ is horizontal because the pressure remains constant to the point of cut off. Then the pressure begins to fall as is represented by the curved line 2′ E. This curve is nearly an equilateral hyperbola.

From Fig, 1 we know that the area B 2′ 2 A is proportional to the work done while the piston moves from A to 2 or during the first half of the stroke. If we use the same data as we did in Fig. 1, the work done must be one-half the work done in the first case, or $\frac{57600}{2} = 28800$ foot-pounds. Also the area B 2′ 2 A is easily found since it is a rectangle. The area 2′ E C 2 is found by dividing it up into small sections, by calculus or by the use of a planimeter.

It is easily seen that the area of the second case Fig. 2, is less than that of Fig. 1. Therefore the work done is less; but the amount of steam admitted is only one-half as much as in the first case.

In the first case, Fig. 1, 8 cubic feet of steam at 50 pounds pressure were admitted per stroke and the work done was found to be 57600 foot-pounds. In the second case only half as much steam is admitted and the work done is $\frac{57600}{2}$ + the amount represented by the area 2′ E C 2. Thus we see that there is a considerable gain by expanding the steam.

Watt's Diagram of Work. Fig. 3 illustrates the method adopted by James Watt to show the action of steam in the cylinder.

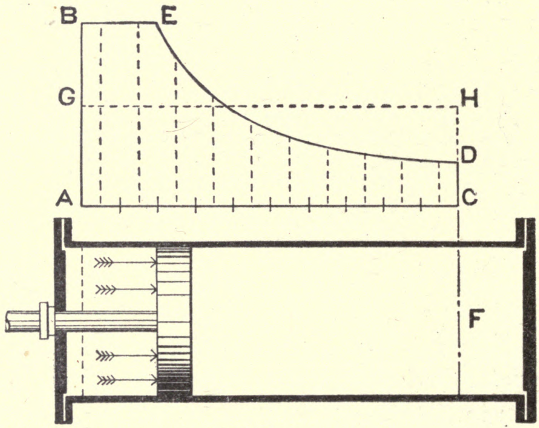


Fig. 3.

The horizontal line A C called the abscissa represents the length of the stroke and is divided into ten equal parts. The vertical line A B called the ordinate indicates the pressure of steam.

When the piston has moved to the point E steam is cut off, that is, a volume of steam equal to $\frac{1}{5}$ the volume of the cylinder expands until it fills the entire cylinder. The area may be found by adding the several pressures (shown by the dotted lines), dividing by the number of divisions, and multiplying by the length.

If by some arrangement of steam tight pistons working in cylinders and having pencils fastened to them, we could get a dia-

gram like that shown in Fig. 3 it would be of great use but too large for convenience.

To obtain the same diagram on a small scale an indicator is used. The value of such a diagram has already been shown when finding the work done in the cylinder. The indicator has enabled engineers to bring the engine of today to its present state of excellence. A correct idea of the action of steam in the cylinder can be obtained only by means of an indicator. It shows whether or not the valves are set properly and how the condenser is working. It also shows the engineer which end of the cylinder is doing the most work. By comparing the expansion line with an equilateral hyperbola, with a curve of constant steam weight, or with an adiabatic curve for steam, the cylinder condensation is calculated.

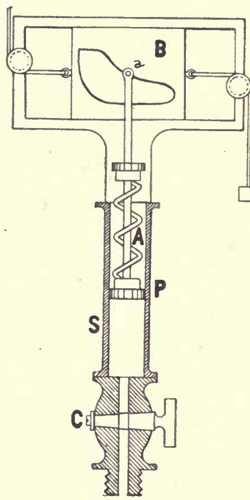


Fig. 4.

James Watt was the first to see the need of accurate knowledge of the action of steam in the cylinder. He invented the indicator. The improved form consisted of a steam cylinder S, about one inch in diameter and six inches long, in which a solid piston P, is accurately fitted. A spiral spring A, is attached to this piston, and controls the motion of a pencil *a*, which is also attached to the piston. This pencil can operate on a sheet of paper fastened to a sliding board, B. This board moves back and forth by means of a weight at one end and a cord at the other which is connected to some reciprocating part of the engine. The indicator cylinder S, may be put in communication with the engine cylinder by means of the cock C. With this instrument a complete diagram can be taken.

Watt's first indicator had no lateral motion, therefore all it showed was the pressure of steam in the cylinder and the perfection of the vacuum.

INDICATORS.

The diagram, or card as it is often called, obtained by the use of an indicator is the result of two motions. The *horizontal* move-

ment of the paper corresponds exactly to the movement of the piston, and the *vertical* movement of the pencil is an exact ratio to that of the pressure of steam in the cylinder. The diagram represents by its *length* the stroke of the engine and by its *height* the steam pressure on the piston at the corresponding point of the stroke. The diagram shows the action of steam on one side of the piston only; to obtain the same information in regard to the other side it is necessary to take another diagram from the other end of the cylinder.

The essential features of an indicator are found in the instrument invented by James Watt. Since his time, however, the many improvements have made the indicator light, compact, durable, and accurate. Watt's diagram was traced on paper stretched on a sliding board but now a revolving drum is used. The height also of Watt's diagram was equal to the movement of the spring, and the pencil arrangement was a simple contrivance. In the indicators of the present day, the spring has a slight movement, the height of the card being obtained by a multiplying arrangement of levers. This method requires a parallel motion to obtain accuracy in the vertical lines; for if a lever is pivoted at one end and power applied near the pivot the lever tends to rise and the free end will describe an arc of a circle, not a straight vertical line.

THE THOMPSON INDICATOR.

Two views of the American Thompson Indicator, the outside and the inside, are shown in Figs. 5 and 6. The form of spring is shown in Fig. 7. The indicator consists of a cylinder in which a piston is fitted, a spring, multiplying lever and parallel motion for the pencil and a cylinder or drum for the paper. The piston, which is .798 inch in diameter = $\frac{1}{2}$ square inch in area, is fitted accurately to the cylinder and has a travel of about one-half inch. When the pressure of steam forces the piston upward it compresses the spring above it; the amount of compression varies with the strength of the spring. The rise of the piston causes the pencil to rise because of the system of levers. The cylinder to which the paper is attached rotates by means of a cord which is fastened to some part of the engine, the crosshead for example.

While the drum revolves and the steam pressure forces the piston to rise, the pencil, lightly touching the paper, describes the diagram.

The parallel movement of this indicator is obtained by a link attached directly to the lever. It is so constructed that there is but little lost motion, hardly any friction and no appreciable error within the limited movement of the pencil.

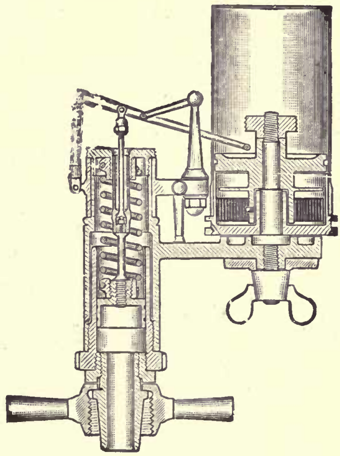


Fig. 5.

The paper cylinder is so constructed that the tension of the coiled spring within the drum may be altered for different speeds of the engine. By this means the cord can be kept taut with little trouble. The cord is led through a hold and kept in contact with the scored wheel by another small one. By this means the cord can be run to any angle. It is convenient to have the cards of about the same size. If we used a spring of such a tension that the pencil would move one inch for every 60 pounds pressure, and there was but 30 pounds pressure in the engine cylinder, the diagram would be but one-half inch high. This diagram would be too small for accurate work. For this reason indicators are provided with sets of springs of varying stiffness which may be used according to the steam pressure. If a spring is of such stiffness that the pencil moves 1 inch for every 50 pounds pressure it is called a 40 pound spring. Others are called 10 pound, 20 pound, 30 pound, etc., springs.

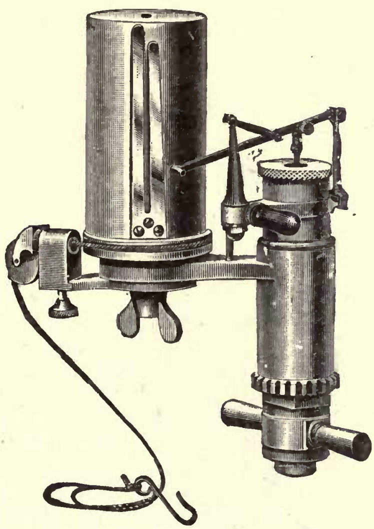


Fig. 6.

To Change the Springs. In selecting the spring for a given pressure, care should be taken that it will easily stand that press-

ure. A safe rule to follow: multiply the scale of the spring by $2\frac{1}{2}$ and subtract 15 for the vacuum.

For example: The maximum pressure for a 50 pound spring is 110 pounds, because $(50 \times 2\frac{1}{2}) - 15 = 125 - 15 = 110$. Springs are made in the following scale: 8, 10, 12, 16, 20, 24, 30, 32, 40, 48, 50, 56, 60, 64, 80, 100. For pressures from 70 to 90 pounds a 40* pound spring should be used, as 80 pounds pressure on a 40 pound spring will raise the pencil 2 inches, and this is a good height for the diagram.

If very high pressures are to be indicated, an extra piston, having an area of $\frac{1}{4}$ square inch, is used. This doubles the allowable pressure on the spring. For instance, if a spring can be used for 110 pounds pressure when the piston is $\frac{1}{2}$ inch in area, it can be used for 220 pounds pressure if the $\frac{1}{4}$ inch piston is used.

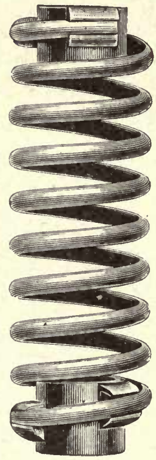


Fig. 7.

When the spring has been selected it is placed in the indicator. First unscrew the milled nut at the top of the steam cylinder and take out the piston with arm and connections. The pencil lever and piston are disconnected by unscrewing the small-headed screw which connects them. If a spring is connected to the piston it should be removed, the selected one substituted and the indicator put together. The spring should always be firmly screwed to the shoulder or the pencil will not properly indicate the pressure.

Care of the Indicator. Before attaching the indicator to the engine cylinder it should be taken apart, cleaned and oiled. If each part works freely and smoothly the spring may be put in and the indicator put together. After connecting to the cylinder, admit steam to it, but do not take cards until it is thoroughly warmed and blows dry steam through the relief. It is not necessary to use lead in connecting as it is likely to get into the indicator. After using, take the indicator apart, clean and oil. Only porpoise or fine watch oil should be used.

THE CROSBY INDICATOR.

The internal arrangement of the parts of the Crosby Indicator

* NOTE: In practice it is better to use a somewhat stiffer spring.

is shown in Fig. 8. The piston, 8, is formed with shallow channels on its outer surface to retain oil which prevents leakage and lubricates the piston.

The socket in the center of the piston is supported by a central web and projects both upward and downward. The upper

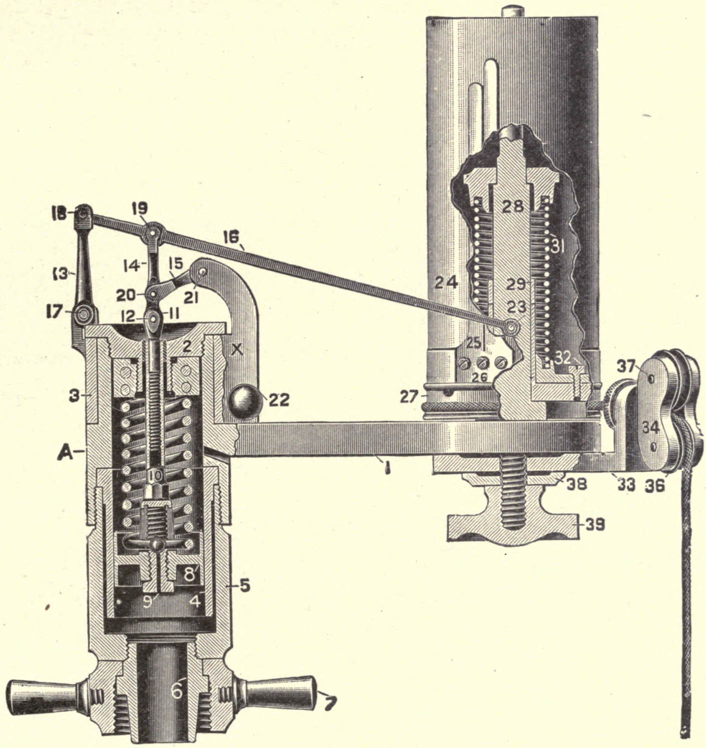


Fig. 8.

portion is threaded inside to receive the lower end of the piston-rod. It has a vertical slot which allows the ball bearing on the end of the spring to drop into a concave bearing on the upper end of the piston-screw 9 which is screwed into the lower part of the socket.

The piston-rod, 10, is made hollow, with the lower end threaded. When the piston-rod is connected to the socket, the former should be screwed into the socket as far as it will go.

The height of the atmospheric line on the diagram depends

upon the amount the swivel head, 11, is screwed into the top of the piston-rod.

A small projection on the lower side of this cap is threaded to screw into the top of the spring and hold it firmly in place. The moving parts are kept in line by this cap. The pencil mechanism is supported by the sleeve 3, which surrounds the upper part of the cylinder; it turns freely and is held in position by the cap.

The pencil mechanism is made as light as is consistent with strength and stiffness. The pencil moves exactly parallel to the piston because the fulcrum of the mechanism and the point of attachment to the piston-rod are always in a straight line. The pencil point moves six times as far as the piston because of the multiplying levers.

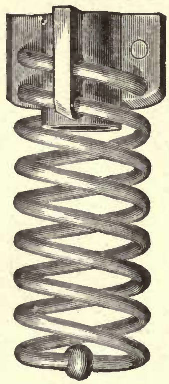


Fig. 9.

The drum 24, is one and one-half inches in diameter, and is rewound when the string is pulled, by a short spiral spring 31.

The piston spring is made of a single piece of steel wire wound from the middle into a double coil. The ends are screwed into a brass head having four radial wings. At the bottom of the spring a small steel bead is firmly attached to the wire. This forms a ball and socket joint with the lower end of the piston-rod. This joint is light and allows the spring to yield to pressure from any direction. These springs are made in the following scale: 8, 12, 16, 20, 24, 30, 40, 50, 60, 80, 100, 120, 150, and 180.

To Insert the Spring. First unscrew the cap 2, then lift the connected parts free from the cylinder by means of the sleeve. The hollow wrench should be held in an inverted position and the piston-rod inserted until the hexagonal part engages the wrench. Then, having the spring shown in Fig. 9 inverted, insert the combined wrench and piston-rod until the steel bead and the end of the spring rests in the concave seat. Now invert the piston and pass the transverse wire at the bottom of the spring through the slot until the threads at the bottom of the piston-rod engage those inside the socket of the piston. With the wrench screw it in as far as it will go.

The piston screw should be loosened slightly before the piston-rod is screwed in, and afterward set up against the bead lightly to prevent lost motion. Then with the sleeve and cap upright, engage the threads of the swivel bead with those inside the piston-rod and screw it up until the lower projection of the cap engages the threads inside the spring top; continue the process until the spring is screwed up firmly against the cap. Holding only by the sleeve 3, turn the piston and the connections until the top of the piston-rod is flush with the shoulder on the swivel head.

Now that the piston and all the connections are in their places, the whole may be inserted in the cylinder and the cap screwed down, which will fix all parts in their proper places. If there is a spring in the cylinder first detach by reversing the above process.

THE TABOR INDICATOR.

The Tabor Indicator is shown in Fig. 10. It is used extensively in the navy. The principle of action and details of construction are similar to the indicators already described; the chief peculiarity being the means employed to obtain a straight line movement for the pencil. Inside the steam cylinder is a lining in which the piston moves. This lining can expand when heated. In the side of the cylinder small holes allow any steam which may leak by the piston to escape. The piston-rod is connected at one end to the piston by means of a ball and socket joint; the other end is connected to the pencil mechanism.

The **pencil mechanism** consists of three pieces, the pencil lever, the back link, and the piston-rod link. The two links are parallel for every position of the pencil. Thus the lower pivots of these links and the pencil point are always in the same straight line. The straight line movement of the pencil is obtained by means of a curved slot in a stationary plate. The pencil bar is provided with a roller which is fitted in such a manner that it can roll from one end of the slot to the other. The curve of the slot guides the pencil bar and is of such a radius that the pencil is caused to move in a straight line. The curve compensates for the tendency of the pencil point to move in an arc of a circle. The

pressure of the pencil on the paper is regulated by a screw which strikes a stop plate attached to the frame. The end of the pencil bar is formed for either a pencil lead or a metallic marking point.

The drum for the paper is made similar to those of other indicators. The backward movement is obtained by a flat spiral spring placed under the drum. The tension of this spring is altered by loosening a thumbscrew, lifting the carriage, and winding or unwinding. A simple pulley guides the driving cord in any direction.

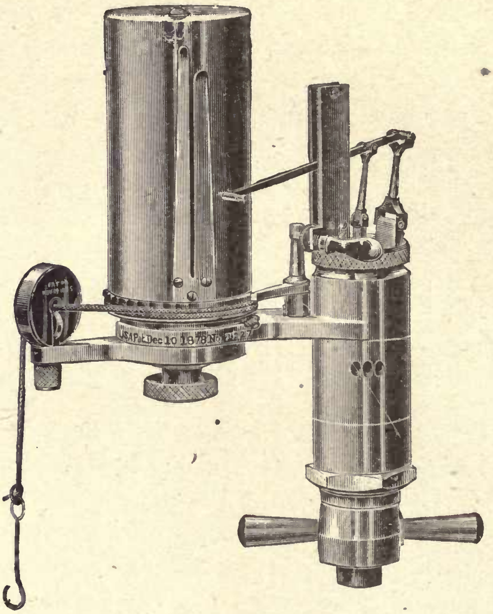


Fig. 10.

The indicator is attached by a coupling having a single thread.

The **springs** of the Tabor indicator are of the duplex type, that is, they are made of two spiral coils of wire. A 50 pound spring is shown in Fig. 11. The wire terminates in fittings at each end. The spring is attached to the upper side of the piston by means of threads cut on the inside of the fitting and on a projection on the piston. The top of the spring is attached to the under side of the cover in a similar manner. The springs are made in the following scales, 8, 10, 12, 16, 20, 24, 30, 32, 40, 48, 50, 60, 64, 80, and 100 pounds. The maximum safe steam pressures (absolute) to which these springs may be subjected are respectively, 10, 15, 20, 24, 40, 48, 70, 75, 95, 112, 120, 140, 152, 180, and 200 pounds.

Change of Location of Atmospheric Line. Unscrew the cap and lift the sleeve and connections from the cylinder. Then turn the piston to the left or right according as the pencil is to be raised or lowered. One revolution causes the pencil to rise $\frac{1}{8}$ inch.

Care of the Indicator. Before attaching the indicator to the engine, steam should be blown through the pipes and cocks so that all particles of dust may be removed. After using, the indicator should be carefully wiped and oiled. The cylinder cap should be unscrewed and all the parts connected to the piston removed. The piston spring and piston-rod should be detached, carefully wiped dry, and then oiled. The inside of the cylinder also should be oiled. Then the piston and piston-rod should be placed in the cylinder and the spring placed in the box. If the indicator has not been used for some time the oil may have become gummed. It may be easily cleaned by wiping with a cloth saturated with naphtha or benzine. It must be oiled again before using. A good test that the indicator is in proper working order is to detach the spring and after replacing the piston and piston-rod, raise the pencil to the highest point. When allowed to fall it should descend to the lowest point freely. The pencil should always have a smooth fine point.

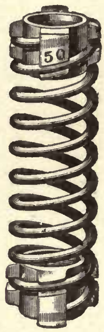


Fig. 11.

To Attach the Indicator to the Engine. Usually all first-class engines are prepared for the indicator before leaving the factory. Holes are drilled and tapped in the cylinder and have plugs screwed in them. These plugs are easily removed and the indicator connections screwed in.

When this is not the case, any engineer can perform the work. Before drilling the holes, in the cylinder, the heads should be removed so that the exact positions of the pistons and the size of the ports and passages may be known. Also with the heads off all chips and particles of dirt from drilling may be easily removed. If it is impossible to remove the heads, a little steam admitted to the cylinder just before the drilling is completed will blow the chips out.

Each end should be drilled and tapped for a one-half inch pipe thread. The holes must be drilled into the clearance space, so that the piston at the ends of the stroke will not cover them. They should also be placed so that currents of steam will not reach them. Before deciding just the points at which to drill the holes, it is well to consider every plan of indicating the engine. The type of engine, the position of the steam chest, the kind of cross-

head, and the position of the eccentric and its connections should all be considered, as well as the most convenient place in the engine room. The holes should not be drilled until the plan shows the proper connections with the reducing motion, convenient access and free passage of steam to the indicator.

When the plan has been adopted, the engine should be placed on dead center, to determine the clearance. The holes should be drilled into the middle of the clearance space.

In common practice for horizontal engines the holes are drilled in the side of the cylinder at each end. Short half inch pipes with quarter upward bends into which the indicator coils may be screwed are inserted in these holes.

It may be more convenient to drill and tap into the top of the cylinder and attach the indicators directly.

For vertical engines, the upper head or cover and the side of the cylinder are often drilled and tapped for the upper and lower indicators respectively. It is preferable to connect the indicators

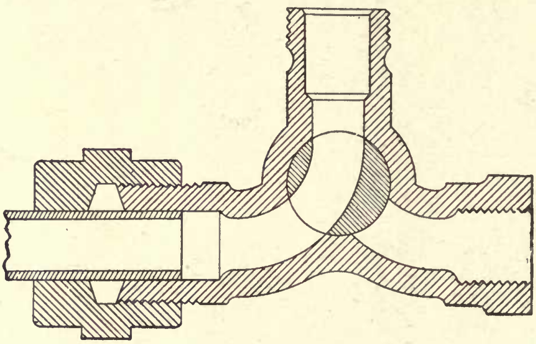


Fig. 12.

to the sides because less pipe and fittings are required and better results obtained.

If only one indicator is to be used for both ends of the cylinder, it may be connected by side pipes and a **three way cock.** By this method both diagrams are taken on the same card and with the loss of but one revolution. Fig. 12 shows the section of a three way cock.

Reducing Motion. As we have already seen the length of the card represents the travel of the piston. As the length of

card is obtained by the rotation of the drum the motion of the drum must be taken from some part of the engine which has a motion coinciding with that of the piston. The crosshead is the most common, reliable and convenient part. The length of the card is much less than the travel of the piston since the stroke is longer than the circumference of the drum so that the movement of the crosshead must be reduced to the length of the diagram.

There are several devices employed to obtain this reduced motion.

A simple form of reducing motion, called the **pantograph,** is shown in Fig. 13. Four links, a, b, c, d, are joined in the form of a parallelogram. One link, a, is prolonged and pivoted at the crosshead C. The point where b and c join is pivoted at the fixed point E. The cord is fastened at D on the link d.

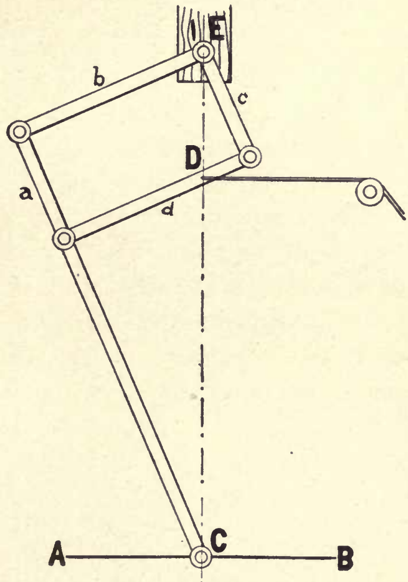


Fig. 13.

The point D must be in the straight line connecting E and C. Then letting A B represent the stroke and h the length of the indicator diagram, we have

$$A\,B : h = E\,C : E\,D, \text{ from which}$$

$$E\,D = \frac{h \times E\,C}{A\,B}$$

Another form of pantograph is shown in Fig. 14. It is placed horizontally with the pivot, B, resting on a support opposite the crosshead when in mid-position. The pivot A, is attached to the crosshead; usually by having the stud A inserted in a hole drilled in the crosshead. If the pivot B is adjusted to the proper height and at the right distance from the crosshead, the cord from the indicator may be attached to the pin E without any pulleys. The

length of the diagram is varied by adjusting the movable bar C D; the pin E must be in the straight line from A to B.

The pantograph is likely to become shaky and loose on account of its many joints. If well made it gives perfect motion.

The reducing motion shown in Fig. 15, called the Brumbo Pulley, is easily and quickly made and can be used on almost any engine. The wooden rod A is usually about twice as long as the stroke. It is pivoted by a bolt or screw at B, a fixed point. At the lower end it is connected by the wooden link, C, to the crosshead.

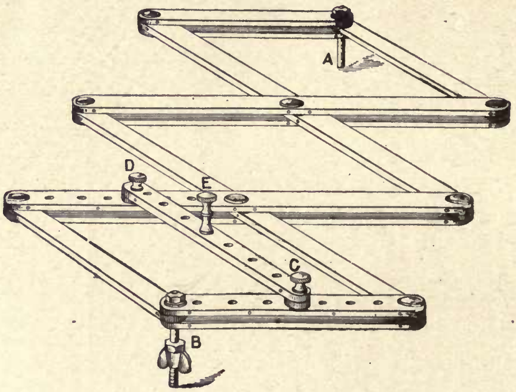


Fig. 14

This link C is usually about one-half the length of the stroke. The sector S may be made either of wood or metal. It should have a groove in the circular edge for the cord, and is made fast to the upper end of the lever A. Its center should coincide with that of the pivot B. The length of the radius of the sector may be found as follows. Divide the length of the lever by the length of the stroke, multiply the result by the length of the desired diagram and the product will be the radius of the sector. For instance, if the lever is 60 inches long and the stroke 30 inches and we wish the diagram to be 3 inches long, the sector should be 6 inches in radius for,

$$\frac{60}{30} \times 3 = 6.$$

To avoid the use of guide pulleys the lever should be hung so that it will swing in a vertical plane parallel with the guides and in line with the indicator. When the crosshead is at mid-stroke,

the lever must be vertical and the point D must be below the axis of the cylinder because it comes above the line at the ends of the stroke.

The reducing lever used for large quick running engines is shown in Fig. 16. The rod A is made of pine wood, tapering toward the lower point and about one inch in thickness. The length is about one and one-half times the length of the stroke. It is suspended by a bolt or screw from some fixed point above the engine, and should swing edgewise and parallel to the guides of the crosshead. The steel stud at the bottom of the rod has a T shaped slot in an iron plate which is attached firmly to the crosshead. The slot should be long enough to retain the stud when the crosshead is at the end of the stroke. To find the point at which the indicator cord should be attached, divide the length of

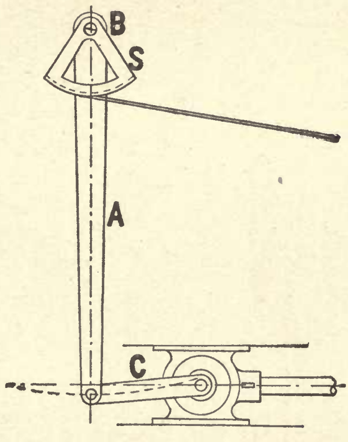


Fig. 15.

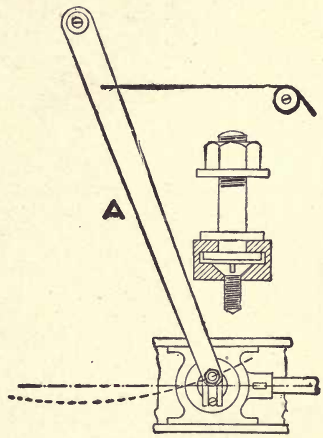


Fig. 16.

the lever by the length of the stroke of the piston and multiply the quotient by the length of the desired diagram. The product is the distance of the point from the pivot at the top of the lever.

Example. A lever is 45 inches long, the piston stroke 30 inches and the diagram to be $3\frac{1}{4}$ inches long. At what distance from the pivot should the indicator cord be attached?

$$\frac{45}{30} \times 3\tfrac{1}{4} = 4\tfrac{7}{8} \text{ inches.}$$

Having placed the indicator in position and obtained the

reducing motion, the length of the cord must be so adjusted that the drum will not strike the stops at either end of the stroke.

For convenience an approximate length of cord is first found and the cord cut in two parts, one attached to the reducing motion; the other to the indicator. A hook should be fastened to the free end of the piece attached to the indicator. A loop is made in the free end of the piece from the reducing motion. The hook is then attached to the loop and the extra length of cord taken up by tying knots. Another method for adjusting the length of the cord is the arrangement shown in Fig. 17. The hook A is attached to the indicator cord. The cord B from the reducing motion passes through the holes in the plate P as shown. To adjust the length of the cord it is slacked at the point B and the plate slipped along the cord.

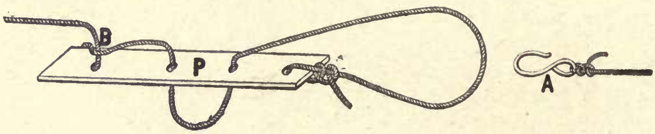


Fig. 17.

To Take Indicator Diagrams. The indicator should be in good working order before attaching to the cylinder; it should be clean, well oiled, and the levers and springs should work smoothly. The pencil point should be sharp and the pressure adjusted to make a distinct fine line.

The spring should be selected that will give a diagram $1\frac{1}{2}$ to 2 inches in height. If the spring chosen is too light, the lines are likely to be wavy from the vibration of the pencil levers.

When the indicator is in position a satisfactory reducing motion obtained and the cord adjusted, the paper should be wrapped smoothly around the drum. The edges projecting over the clips should be folded back so that they will not touch the pencil lever. Before taking the card, allow the steam to enter the indicator and move the piston up and down until the parts have become thoroughly warmed. Then pass the pencil against the paper long enough to take the diagram. Some engineers allow the pencil to remain in contact with the paper during but one revolution of the engine; others trace the diagram two or

more times. After the diagram is taken the cock is shut and without unhooking the cord the pencil is again pressed to the paper to take the atmospheric line. The cord is then disconnected and the card removed from the drum. The scale of spring, the dimensions and speed of the engine, the date, and all useful particulars are written on the cards.

If one indicator is used for both ends the three way cock shown in Fig. 12 is opened to admit steam from one end, the diagram taken; then opened for the other end and that diagram taken. Then the steam is shut off from both ends and the atmospheric line taken.

As has been said before, the indicator is of great importance to the engineer. It is used to find the indicated horse-power of the engine, and by comparison of the indicated horse-power with the brake horse-power, the **mechanical efficiency** is obtained. The indicator card shows several other things; the *time* and *manner* of the **four events** of the stroke, namely, the admission, cut-off, release, and compression. These four events make up what is called the **steam distribution.** It shows faults in the setting and working of the valves.

We have seen how the indicator diagram represents the net work done on the piston in one stroke.

Work is equal to pressure multiplied by the distance through which it acts. The *distance* is the length of stroke multiplied by the number of strokes per minute. The *pressure* is the *average net* pressure acting on the piston during the stroke. This average net pressure is called the **mean effective pressure.** If we know from the indicator card the mean effective pressure per square inch, we can find the total pressure by multiplying it by the area of the piston in square inches.

The distance per minute is equal to the length of stroke multiplied by the number of strokes.

Let P = mean effective pressure in pounds per square inch.
A = area of piston in square inches.
L = length of stroke in feet.
N = number of strokes per minute.

Then the work done per minute,

$$W = P\,L\,A\,N.$$

Since one horse-power is the rate of doing work when 33,000 foot-pounds of work are done per minute the **indicated horse-power** of an engine is obtained by means of the formula,

$$H.\ P. = \frac{P\ L\ A\ N}{33{,}000}$$

The length of the stroke, the area of the piston and the number of strokes are easily found. Then all that remains to be determined before the horse-power is calculated is the value of P, or the mean effective pressure.

Suppose one side of the piston is in communication with the boiler during the entire stroke, the mean pressure is then the boiler pressure. But if the supply of steam is cut off before the stroke is completed, the mean pressure will not equal the boiler pressure. In Fig. 1 we saw that the area of the shaded portion was equal to the length multiplied by the height. The area of a figure of any shape can be reduced to that of a rectangle having a length equal to the extreme length of the figure. Then whenever we know the area and length, we can find the height or mean height by dividing the area by the length.

Suppose a diagram like that shown in Fig. 2 has an area of $5\frac{1}{4}$ square inches and its extreme length is $3\frac{1}{2}$ inches; then the height is 5.25 divided by $3.5 = 1.5$ inches. Then with any indicator card the area of the diagram is equal to the area of a rectangle, the length of which is known and the height can easily be computed.

Suppose that we have taken a card and know that the mean height is $1\frac{1}{2}$ inches. In order to find the horse-power we must reduce the $1\frac{1}{2}$ inches to pounds pressure. If we multiply the height by the strength of the spring we get the desired result. For instance, if we had used a 30 pound spring (that is one which causes the pencil of the indicator to move one inch for every 30 pounds pressure in the cylinder) the height $1\frac{1}{2}$ inches would equal $30 \times 1\frac{1}{2} = 45$ pounds pressure.

Example. An indicator card has an area of 1.925 square inches and is 2.2 inches long. If a 60 pound spring is used what is the mean pressure?

The mean height equals $\frac{1.925}{2.2} = .875$ inch and

$$.875 \times 60 = 52.5 \text{ pounds.}$$

Suppose the engine from which the above card was taken had a piston 14 inches in diameter and a stroke of 24 inches. If it were running at 150 revolutions per minnte what is its horse-power? Assume the mean effective pressure to be the same for both sides of the piston.

$$\text{H. P.} = \frac{P\,L\,A\,N}{33{,}000}$$
$$= \frac{52.5 \times 2 \times 153.94 \times 300}{33{,}000}$$
$$= 147. \text{ (about)}$$

In most engines more work is done at one end of the cylinder that at the other; it is not safe then to assume the mean effective pressure of one side the same as that of the other. Cards should be taken from each end and calculated for mean effective pressure separately, then averaged. Also the area of one side of the piston is greater than the other on account of the piston-rod. The two ends may be figured separately or the average area of the two sides of the piston may be used as the value of A.

Another method is to find the work done at each end of the cylinder and then add the results. This enables the engineer to know if his valves are set so that each end does about the same amount of work.

An engine has the following dimensions. The piston is 12 inches in diameter, the piston-rod is $2\frac{1}{8}$ inches in diameter and the length of stroke is 34 inches. While running at 92 revolutions cards were taken. The area of the card from the head end was 5.36 square inches, that of the crank end was 5.30 square inches, and a 40 pound spring was used. The cards were 3.72 inches long. We wish to know what horse-power the engine developed and which end was doing the most work.

The area of the piston is 113.097 square inches for the head end and 113.097 — 3.5466 = 109.55 square inches for the crank end.

Then for the head end,

$$\text{H. P.} = \frac{P\,L\,A\,N}{33{,}000} = \frac{P \times 34 \times 113.097 \times 92}{12 \times 33{,}000} = .8933\ P$$

and for the crank end,

$$\text{H. P.} = \frac{P\,L\,A\,N}{33{,}000} = \frac{P \times 34 \times 109.55 \times 92}{12 \times 33{,}000} = .8653\ P$$

The value of P is found from the cards. Since the cards were 3.72 inches long and the area of the card from the head end was 5.36 square inches the mean ordinate is $\frac{5.36}{3.72} = 1.44$ inches which equals $1.44 \times 40 = 57.6$ pounds. The horse-power would be,

$$.8933 \times 57.6 = 51.45$$

For the crank end the mean effective pressure is found as before,

$$\frac{5.30}{3.72} = 1.425 \text{ and } 1.425 \times 40 = 57.$$

The horse-power would be,

$$.8653 \times 57 = 49.32$$

The horse-power is evidently the sum of these two quantities or $51.45 + 49.32 = 100.77$.

The head end is doing more work than the crank end but the difference is slight being only,

$$51.45 - 49.32 = 2.13 \text{ horse-power,}$$

or about 2.1 per cent. of the total power.

Considerable arithmetical work is necessary when the I. H. P. is found from the formula,

$$\text{I. H. P.} = \frac{\text{PLAN}}{33{,}000}$$

and the chances for error are of course, great. To save time and reduce the chance for error a table of engine constants has been prepared. The number of strokes, or twice the number of revolutions, multiplied by the length of stroke in feet is called the **piston speed.** Then in the formula I. H. P. $= \frac{\text{PLAN}}{33{,}000}$, L N = piston speed in feet per minute. In the following table, the I. H. P. of an engine is easily computed by multiplying the constant, corresponding to the diameter of the piston, by the piston speed and by the M. E. P. Or, in other words, the constants in the table equal the horse-power for an engine with a given diameter of piston having a piston speed of one foot per minute and a M. E. P. of one pound.

TABLE OF ENGINE CONSTANTS.

Diameter of Cylinder.	Even Inches.	+ 1/8 or .125.	+ 1/4 or .25.	+ 3/8 or .375.	+ 1/2 or .5.	+ 5/8 or .625.	+ 3/4 or .75.	+ 7/8 or .875.
1	.0000238	.0000301	.0000372	.0000450	.0000535	.0000628	.0000729	.0000837
2	.0000952	.0001074	.0001205	.0001342	.0001487	.0001640	.0001800	.0001967
3	.0002142	.0002324	.0002514	.0002711	.0002915	.0003127	.0003347	.0003574
4	.0003808	.0004050	.0004299	.0004554	.0004819	.0005091	.0005370	.0005656
5	.0005950	.0006251	.0006560	.0006876	.0007199	.0007530	.0007869	.0008215
6	.0008568	.0008929	.0009297	.0009672	.0010055	.0010445	.0010844	.0011249
7	.0011662	.0012082	.0012510	.0012944	.0013387	.0013837	.0014295	.0014759
8	.0015232	.0015711	.0016198	.0016693	.0017195	.0017705	.0018222	.0018746
9	.0019278	.0019817	.0020363	.0020916	.0021479	.0022048	.0022625	.0023209
10	.0023800	.0024398	.0025004	.0025618	.0026239	.0026867	.0027502	.0028147
11	.0028798	.0029456	.0030121	.0030794	.0031475	.0032163	.0032859	.0033561
12	.0034272	.0034990	.0035714	.0036447	.0037187	.0037934	.0038690	.0039452
13	.0040222	.0040999	.0041783	.0042576	.0043375	.0044182	.0044997	.0045819
14	.0046648	.0047484	.0048328	.0049181	.0050039	.0050906	.0051780	.0052661
15	.0053550	.0054446	.0055349	.0056261	.0057179	.0058105	.0059039	.0059979
16	.0060928	.0061884	.0062847	.0063817	.0064795	.0065780	.0066774	.0067774
17	.0068782	.0069797	.0070819	.0071850	.0072887	.0073932	.0074985	.0076044
18	.0077112	.0078187	.0079268	.0080360	.0081452	.0082560	.0083672	.0084791
19	.0085918	.0087052	.0088193	.0089343	.0090499	.0091663	.0092835	.0094013
20	.0095200	.0096393	.0097594	.0098803	.0100019	.0101243	.0102474	.0103712
21	.0104958	.0106211	.0107472	.0108739	.0110015	.0111299	.0112589	.0113886
22	.0115192	.0116505	.0117825	.0119152	.0120487	.0121830	.0123179	.0124537
23	.0125902	.0127274	.0128654	.0130040	.0131435	.0132837	.0134247	.0135664
24	.0137088	.0138519	.0139959	.0141405	.0142859	.0144321	.0145789	.0147266
25	.0148750	.0150241	.0151739	.0153246	.0154759	.0156280	.0157809	.0159345
26	.0160888	.0162439	.0163997	.0165563	.0167135	.0168716	.0170304	.0171899
27	.0173502	.0175112	.0176729	.0178355	.0179988	.0181627	.0183275	.0184929
28	.0186592	.0188262	.0189939	.0191624	.0193316	.0195015	.0196722	.0198436
29	.0200158	.0201887	.0203624	.0205368	.0207119	.0208879	.0210645	.0212418
30	.0214200	.0215988	.0217785	.0219588	.0221399	.0223218	.0225044	.0226877
31	.0228718	.0230566	.0232422	.0234285	.0236155	.0238033	.0239919	.0241812
32	.0243712	.0245619	.0247535	.0249457	.0251387	.0253325	.0255269	.0257222
33	.0259182	.0261149	.0263124	.0265106	.0267095	.0269092	.0271097	.0273109
34	.0275128	.0277155	.0279189	.0281231	.0283279	.0285336	.0287399	.0289471
35	.0291550	.0293636	.0295729	.0297831	.0299939	.0302056	.0304179	.0306309
36	.0308448	.0310594	.0312747	.0314908	.0317075	.0319251	.0321434	.0323624
37	.0325822	.0328027	.0330239	.0332460	.0334687	.0336922	.0339165	.0341415
38	.0343672	.0345937	.0348209	.0350489	.0352775	.0355070	.0357372	.0359681
39	.0361998	.0364322	.0366654	.0368993	.0371339	.0373694	.0376055	.0378424
40	.0380800	.0383184	.0385575	.0387973	.0390379	.0392793	.0395214	.0397642
41	.0400078	.0402521	.0404972	.0407430	.0409895	.0412368	.0414849	.0417337
42	.0419832	.0422335	.0424845	.0427362	.0429887	.0432420	.0434959	.0437507
43	.0440062	.0442624	.0445194	.0447771	.0450355	.0452947	.0455547	.0458154
44	.0460768	.0463389	.0466019	.0468655	.0471299	.0473951	.0476609	.0479276
45	.0481950	.0484631	.0487320	.0490016	.0492719	.0495430	.0498149	.0508875
46	.0503608	.0506349	.0509097	.0511853	.0514615	.0517386	.0520164	.0522949
47	.0525742	.0528542	.0531349	.0534165	.0536988	.0539818	.0542655	.0545499
48	.0548352	.0551212	.0554079	.0556953	.0559835	.0562725	.0565622	.0568526
49	.0571438	.0574357	.0577284	.0580218	.0583159	.0586109	.0589065	.0592029
50	.0595000	.0597979	.0600965	.0603959	.0606959	.0609969	.0612984	.0616007
51	.0619038	.0622076	.0625122	.0628175	.0632235	.0634304	.0637379	.0640462
52	.0643552	.0646649	.0649753	.0652867	.0655987	.0659115	.0662250	.0665392
53	.0668542	.0671699	.0674864	.0678036	.0681215	.0684402	.0687597	.0690799
54	.0694008	.0697225	.0700449	.0703681	.0705293	.0710166	.0713419	.0716681
55	.0719950	.0724226	.0726510	.0729801	.0733099	.0736406	.0739719	.0743039
56	.0746368	.0749704	.0753047	.0756398	.0759755	.0763120	.0766494	.0769874
57	.0773262	.0776657	.0780060	.0783476	0786887	.0790312	.0793745	.0797185
58	.0800632	.0804087	.0807549	.0811019	.0814495	.0817980	.0821472	.0824971
59	.0828478	.0831992	.0835514	.0839043	.0842579	.0846123	.0849675	.0853234
60	.0856800	.0860374	.0863955	.0867543	.0871139	.0874743	.0878354	.0881973

To Use the Table. If the diameter of the piston is an even number, the constant is found in the second column; if it contains a fraction the constant is found by following the column horizontally until the required fraction is reached. The constant multiplied by the piston speed in feet per minute and by the M. E. P. in pounds per square inch gives the I. H. P.

Example. An engine runs at 75 revolutions. The stroke is 4 feet; if the M. E. P. is 48 pounds and the piston $27\frac{3}{8}$ inches in diameter what is the I. H. P.

From the table the constant for a piston $27\frac{3}{8}$ inches in diameter is .0178355. The piston speed is $150 \times 4 = 600$ feet per minute. Then the I. H. P. is,

$$.0178355 \times 600 \times 48 = 513.66$$

The horse-power as above calculated is called the indicated horse-power and is usually written I. H. P. Although the above calculation shows the amount of power the engine develops it does

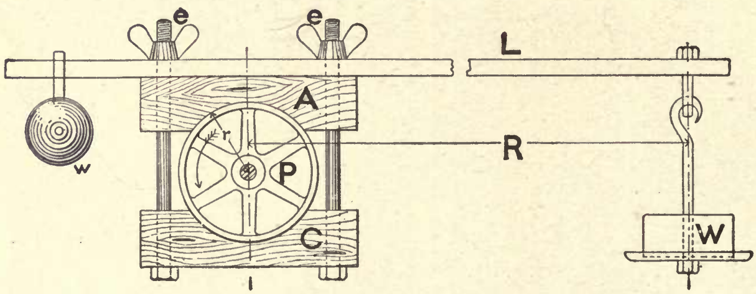


Fig. 18.

not show the *available* power since part of the indicated horse-power is used to run the engine itself, that is, to overcome the friction of the parts. To determine how much power can be used to run machinery some form of absorption dynamometer or friction brake is attached to the engine. The power thus obtained is called the **Brake Horse Power** or B. H. P. It is more satisfactory for both the owner and builder to know the B. H. P. than to know the I. H. P.

The Prony Brake, Fig. 18, is one of the simplest absorption dynamometers. The two wooden blocks A and C are held together against the rim of the pulley P by bolts. The thumb-nuts, *e*, *e*, being

used to adjust the pressure. By means of the bolts the arm L is held to the upper block. From this arm is suspended the ball weight, w, which by sliding along the arm counterbalances the weight of the arm and pan at the other end. The pulley revolves at the required speed in the direction indicated by the arrow. The bolts are tightened until the lever remains stationary in a horizontal position when a known weight, W, is hung at the end.

The amount of work absorbed by the brake depends upon the weight W, the length, R, and the speed. It is independent of the diameter of the pulley and the pressure of the block because the moments of force about the center of the pulley are equal when the lever L, is horizontal. Letting f equal the co-efficient of friction, p the pressure of the blocks and r the radius of the pulley,

$$f\,p\,r = \mathrm{W\,R}\ .$$

The work done at the face of the pulley equals the force multiplied by the distance or the pressure multiplied by the number of feet passed through.

Let N = the number of revolutions per minute. Then the distance passed through per minute equals $2\,\pi\,r\,\mathrm{N}$ and the work done equals $2\,\pi\,r\,\mathrm{N}\,f\,p$. Then as $f\,p\,r = \mathrm{W\,R}$, the work done at the rim of the pulley equals the left hand side of the equation multiplied by $2\,\pi\,\mathrm{N}$, and to keep both sides equal we multiply W R by $2\,\pi\,\mathrm{N}$. Hence the work done is expressed by the formula

$$2\,\pi\,\mathrm{N\,W\,R} \text{ and,}$$

$$\text{B. H. P.} = \frac{2\,\pi\,\mathrm{N\,W\,R}}{33{,}000}$$

$$= .0001904\ \mathrm{N\,W\,R}$$

A Prony brake with an arm 4 feet long was attached to the pulley on the fly wheel of an engine. The weight in the scale pan was 50 pounds and the speed of the engine 300 revolutions. Find the brake horse power.

$$\text{B. H. P.} = .0001904 \times 300 \times 50 \times 4$$

$$= 11.424$$

The **rope brake** shown in Fig. 19 is easily constructed of material at hand and being self-adjusting needs no accurate fitting. For large powers the number of ropes may be increased. It is considered a most convenient and reliable brake. In Fig. 19 the spring

balance, B, is shown in a horizontal position. This is not at all necessary; if convenient the vertical position may be used. The ropes are held to the pulley or fly-wheel face by blocks of wood, O. The weights at W may be replaced by a spring balance if desirable.

To calculate the Brake Horse Power, subtract the pull registered by the spring balance, B, from the load at W. The lever arm is the radius of the pulley plus $\frac{1}{2}$ the diameter of the rope. The formula is,

$$\text{B. H. P.} = \frac{2\,\pi\,R\,N\,(W - B)}{33{,}000}$$
$$= .0001904\,R\,N\,(W - B)\,*$$

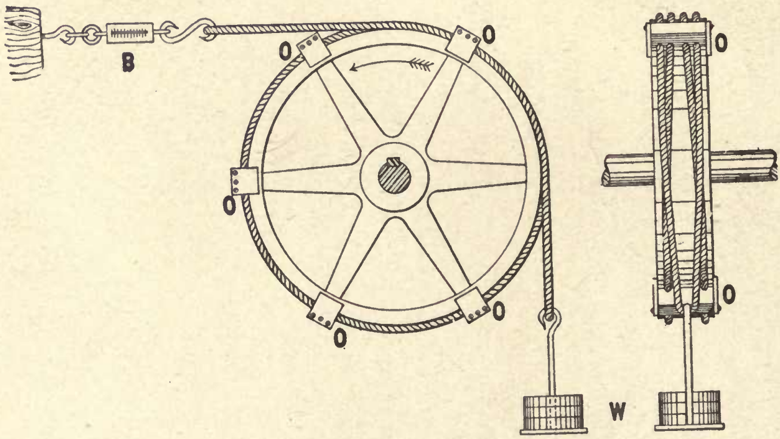


Fig. 19.

Example. A rope brake is attached to a gas engine. The average reading of the spring balance is 8 pounds. W = 80 pounds. If the radius of the brake wheel is 28 inches and the rope 1 inch in diameter, what is the B. H. P. when the engine makes 350 revolutions per minute?

$$R = 28 + \tfrac{1}{2} = 28\tfrac{1}{2} \text{ inches} = \tfrac{28.5}{12} \text{ feet}$$
$$\text{B. H. P} = .0001904\,R\,N\,(W - B)$$
$$= .0001904 \times \tfrac{28.5}{12} \times 72 \times 350$$
$$= 11.4 \text{ Ans}$$

If both the indicated horse-power and the brake horse-power

* NOTE: If B is greater than W, the engine is running in the opposite direction. Use the formula B. H. P. = .0001904 R N (B — W).

are known the power used in friction is found by subtracting the B. H. P. from the I. H. P.

The **mechanical efficiency** of the engine is the **ratio** of the B. H. P. to the I. H. P. or,

$$E = \frac{B.\ H.\ P.}{I.\ H.\ P.}$$

If an engine of 18.2 indicated horse-power develops at a trial 16.02 brake horse-power, what is its mechanical efficiency?

$$E = \frac{B.\ H.\ P.}{I.\ H.\ P.}$$

$$= \frac{16.02}{18.2} = .88$$

$$= 88\ \%\ \text{Efficiency.}$$

Brakes should be well lubricated. For small powers the heat generated by friction between the ropes or blocks and the rim of the wheel, will be conducted away by radiation but for large powers some additional means is necessary. In case there are flanges on the wheel, water can be introduced into the wheel, the flanges keeping it from flowing out and centrifugal force keeping it in contact with the rim. The amount of water can be regulated so that all may be evaporated, or a scoop can be arranged to carry off the water. In all cases the water should flow continuously.

To Find the Area of Cards. M. E. P. or the mean effective pressure is equal to the area of the indicator diagram divided by the length. The length is easily found by measurement but to find the area is more difficult since the shape is irregular. If the figure were regular its area could be found by geometry or by simple formulas.

The area of the indicator card can be found in two ways. By dividing the diagram into sections and by the use of a planimeter. The former is only an approximate method; the area thus found is nearly correct if the number of divisions is great.

Tangents at each end, perpendicular to the atmospheric line are first drawn. The horizontal distance between these tangents is then divided into 10 or more equal parts. The horizontal length of each section is then divided into two equal parts and lines perpendicular to the atmospheric line drawn through these points of

division. The sum of the lengths of all these lines is divided by the number of lines to get the average. This average length or average ordinate multiplied by the scale of spring gives the mean effective pressure.

Fig. 20 is the card from the crank end of an engine. The

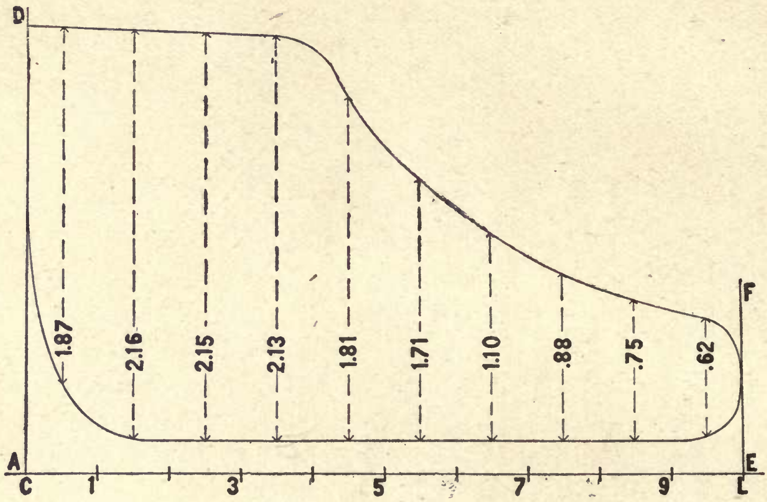


Fig. 20.

line C L is the atmospheric line and the lines A D and E F are drawn perpendicular to it and tangent to the extreme ends of the diagram. The line A E is divided into 10 equal parts and lines

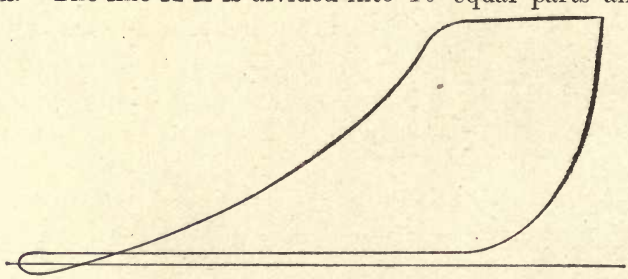


Fig. 21.

are drawn through points marking the centers of the divisions. On each of these lines the length is marked. The sum of the lengths is 15.18 and 15.18 divided by 10 is 1.518. If the scale of spring is 40 pounds, 1.518 multiplied by 40 is the M. E. P. or 60.7 = M. E. P.

The horizontal length may be divided into any number of equal parts but 10 or 20 makes the computation easy. The operation of finding the M. E. P. for the head end is exactly the same. The average M. E. P. for one revolution of the engine is the average of the two mean effective pressures.

In case the diagram is very irregular it should be divided into 20 equal parts instead of 10. If there is a loop in the diagram as shown in Fig. 21 the area of the loop must be subtracted

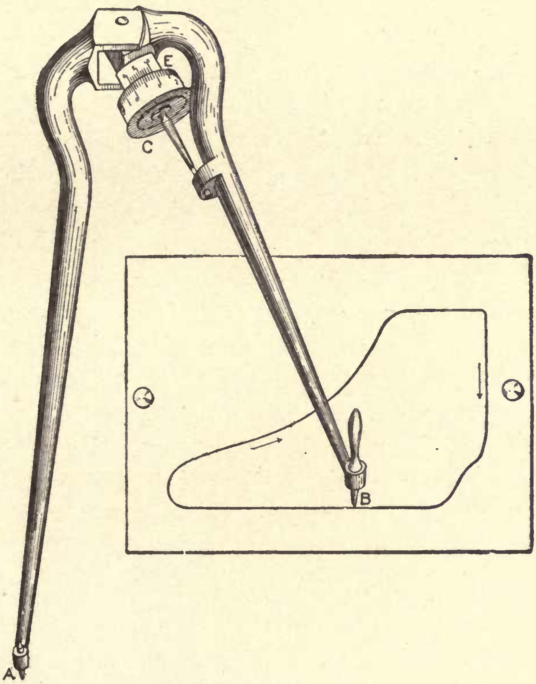


Fig. 22.

from the area of the other part as it represents work done by the piston on the steam and therefore loss.

The lengths may be marked off on a piece of paper if a good scale is not at hand.

A more accurate result is obtained by using an instrument called the **planimeter.** There are several planimeters and averaging instruments in common use for determining the mean effective pressure of indicator cards. The planimeter shown in Fig.

22 is one of the most simple and is called the Amsler Polar Planimeter from its inventor Prof. Amsler. The cut is about one-half the size of the instrument. It consists of two arms free to move about a pivot and a roller graduated in inches and tenths of inches. A vernier is placed with the roller so the areas may be read in hundredths of a square inch. The point A is kept stationary and the tracer B is moved once around the outline of the diagram. The area in square inches of the diagram is read from the roller C and the vernier E.

To Use the Planimeter. The diagram should be fastened to some flat unglazed surface, such as a drawing board, by means of thumb tacks, springs or pins. The point A is pressed into the paper so that it will hold in place. The point B is set at any point in the outline of the diagram and the roller set at zero. Follow the outline of the diagram carefully in the direction of the hands of a watch as indicated by the arrows in Fig. 22 until the tracer has moved completely around the diagram. The result is then read to hundredths of an inch from the roller. Suppose after tracing over the outline we find that the largest figure that has passed the zero of the vernier is 3; the number of graduations (tenths) that have passed the zero to be 5 and the number (hundredths) of the graduations in the roller that exactly coincides with a graduation on the vernier to be 9. Then the area is 3.59 square inches.

Often at the start the roller is not adjusted so that the zeros coincide but the reading is taken and subtracted from the final reading. Thus if the first reading is 4.63 and the second 7.31 the area is 7.31 — 4.63 = 2.68 square inches. In case the second reading is less than the first, add 10 to the second reading then subtract.

This instrument is very valuable to an engineer who takes indicator cards. The results obtained are very accurate, the error being small. Ten or twelve diagrams can be measured by this instrument in the same time that is necessary to measure a single card by the method of ordinates.

It is well to run over the area three or four times and take an average as the tracing of the diagram cannot be absolutely correct at any time.

THERMAL EFFICIENCY.

The **thermal efficiency** of the steam engine is found in the same manner as that of any other heat engine. The efficiency depends upon the limits of temperature and not upon the nature of the working medium.

Let T_1 = absolute temperature of the heat received by the engine.
T_2 = absolute temperature of the heat rejected by the engine.
E = efficiency of engine.

Then,

$$E = \frac{T_1 - T_2}{T_1}$$

or, the efficiency equals the temperature of the heat rejected, subtracted from the temperature of the heat received and the result divided by the temperature of the heat received.

Suppose an engine is supplied with steam at 120 pounds absolute pressure and the exhaust is atmospheric pressure. What is the efficiency?

The absolute temperature corresponding to 120 pounds absolute pressure is 341.05° + 461° and the temperature of atmospheric pressure is 212° + 461°.

Then,

$$E = \frac{802.05 - 673}{802.05} = .16 \text{ or } 16 \text{ per cent.}$$

If the engine had been of the condensing type and the exhaust pressure one pound above the vacuum, the efficiency would be as follows:

The temperature of one pound absolute pressure is 101.99° + 461°.

$$E = \frac{802.05 - 562.99}{802.05} = .30 \text{ or } 30 \text{ per cent.}$$

In actual engines this efficiency cannot be obtained because the difference between the amount of heat received and that rejected is not all converted into work. Part of it is lost by radiation, conduction, leakage, etc. Also cylinder condensation reduces the efficiency.

The Theoretical Indicator Diagram. An indicator diagram is the result of two movements; a horizontal movement of the

paper and a vertical movement of the pencil. The horizontal movement exactly corresponds to the movement of the piston of the engine and the vertical movement exactly corresponds to the pressure of steam in the cylinder.

The shape of the indicator card depends upon the manner in which steam is admitted to and released from the cylinder. Different engines give different shaped indicator cards and the cards taken from an engine vary with the conditions. Figs. 1 and 2 show theoretical indicator cards from a non-condensing engine without clearance; the former being for the case that has admission during the whole stroke. The diagram of Fig. 2 shows the cut off at $\frac{1}{2}$ stroke. All practical engines have clearance and slight compression; so the theoretical diagram assumes the shape shown in Fig. 23. In this card the admission line H A is vertical, the steam line A C is horizontal, the expansion line C D an hyperbolic curve, the exhaust line D B vertical, the back pressure line B F horizontal and the compression curve an hyperbola.

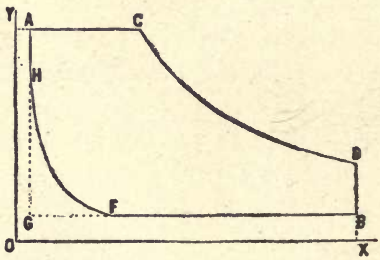


Fig. 23.

The **actual** shape is somewhat different from the **theoretical** mainly because the valves do not open and close quickly, the ports offer some resistance to the passage of the steam and the back pressure is neither atmospheric in the non-condensing engine nor absolute vacuum in the condensing engine.

The diagram shown in Fig. 24 is a practical diagram and like those taken from engines.

The **atmospheric line** L M is the line drawn by the pencil of the indicator when the connection to the engine is closed and both sides of the piston of the indicator are open to the atmosphere. It is the zero of the steam gage.

The **admission line** H A shows the rise of pressure due to the admission of steam to the cylinder. If the steam is admitted quickly when the engine is nearly on dead center this line will be very nearly vertical.

The **steam line** A C is drawn while the valve admits steam

to the cylinder. This line is horizontal if there is no wire-drawing.

The **point of cut off** C, indicates the point at which the admission of steam is stopped by the closing of the valve. This point is rounding since the valve closes slowly. Sometimes it is difficult to determine the exact point where cut off takes place; it is usually where the curve changes from concave to convex.

The **expansion curve** C D shows the fall in pressure as the steam expands while the piston moves toward the end of the stroke.

The **point of release** D shows the point at which the exhaust

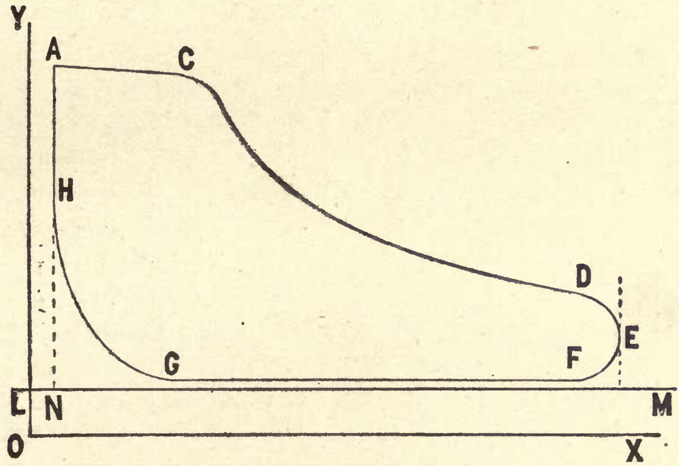


Fig. 24.

valve opens. The rounding is due to the slow action of the valve when opening. Because of this slow action of the valve, release begins a little before the end of the forward stroke.

The **exhaust line** D E F represents the loss in pressure which occurs while the valve opens to exhaust at and near the end of the stroke.

The **back pressure line** F G shows the back pressure against which the piston acts during the return stroke. For a condensing engine this line is below the atmospheric line L M, the distance below being dependent upon the state of the vacuum in the condenser. For cards taken from a non-condensing engine the back pressure line is a little above the atmospheric line.

The **point of exhaust closure** G is the point where the valve

closes to exhaust. The exact point is not clearly defined as the curve shows a change of pressure due to the gradual closing of the valve.

The **compression curve** G H shows the rise of pressure due to the compression of the steam remaining in the cylinder after the valve has closed to exhaust.

The **zero line** of pressure or line of absolute vacuum O X is drawn below and parallel to the atmospheric line. The distance between the lines O X and L M represents 14.7 pounds pressure.

The **clearance line** O Y is drawn perpendicular to the line of absolute vacuum and at a distance from the end of the diagram

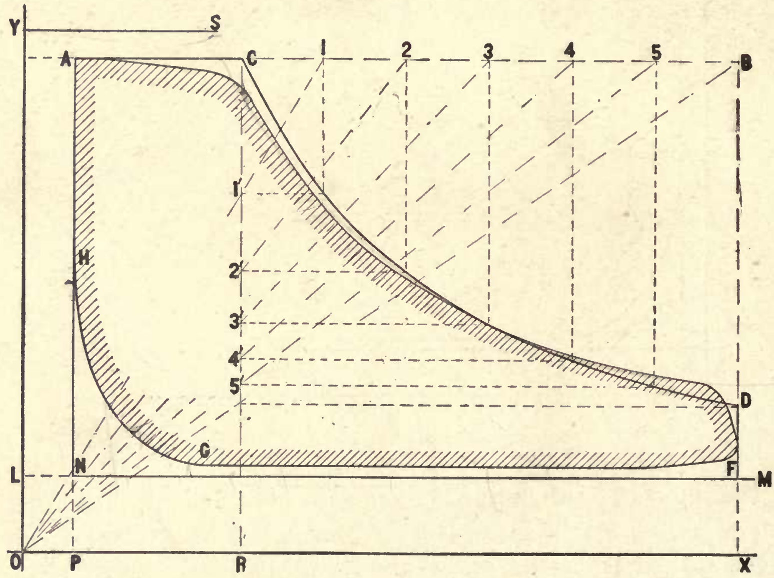


Fig. 25.

equal to the same per cent. of the length of the diagram as the clearance volume is of the piston displacement, or

$$\frac{\text{L N}}{\text{L M}} = \frac{\text{clearance volume}}{\text{volume of cylinder}}.$$

It is readily seen that the area of an actual indicator diagram is less than that of a theoretical card. This is because of the round corners at cut off and exhaust, the back pressure and the compression. Sometimes it is useful, especially in designing engines, to draw the theoretical indicator card.

To Draw the Theoretical Card. To draw an ideal diagram draw P X equal to the length of stroke and O P equal to the clearance shown in Fig. 25. Draw O Y and P A perpendicular to O X and draw Y S parallel to O X and at a height corresponding to the boiler pressure.

The line of initial pressure A C is then drawn parallel to Y S and is usually taken as from 90 to 95 per cent. of the boiler pressure if there is no special cause for loss. Then take A C as the portion of the stroke at which steam is admitted so that $\frac{P\,X}{A\,C}$ = the ratio of expansion. The expansion line is considered a hyperbolic curve with O Y and O X as asymptotes. To draw the hyperbolic curve. First draw the line A C B parallel to the atmospheric line and F D B and R C perpendicular to it. Then make points 1, 2, 3, 4, etc., on C B and connect them with the point O. At the points 1′, 2′, 3′, 4′, etc., where these lines intersect the line R C draw parallels to C B until they meet perpendiculars from 1, 2, 3, 4. The point of intersection of these lines are points in the hyperbolic curve C D, as shown in Fig. 25. Any number of points may be used; but there must be enough to determine the curve.

The area A C D F N H is the theoretical card, with a given boiler pressure, ratio of expansion and an assumed back pressure. The actual card for the same data would be more nearly like the shaded area which lies mostly within the outline of the theoretical card. In designing engines it is well to know the ratio of the actual to the ideal card for all types of engines.

This ratio varies with the speed, type of engine, and kind of valves and has the following values.

For ordinary plain slide valve engines,	.8 to .9
For small engines, about,	.85
For large engines, about,	.90
For engines with high speed, about,	.75
For compound engines,	.75 to .80
For compound engines, high speed,	.65 to .75
For triple expansion engines,	.60 to .70
For triple expansion engines, high speed,	.50 to .60

CARDS FOR COMPOUND ENGINES.

In Fig. 26 the ideal cards of a **tandem** compound engine of the Woolf type are shown. The diagram A C D E F G H is from the high pressure cylinder and the diagram L M N P Q R from the low pressure. S T is the atmospheric line.

The line H A is the admission line, A C the steam line, and C D the expansion line. These lines are similar to those of a single cylinder engine diagram. At D, the point of release, the pressure drops slightly as steam is admitted to the low pressure

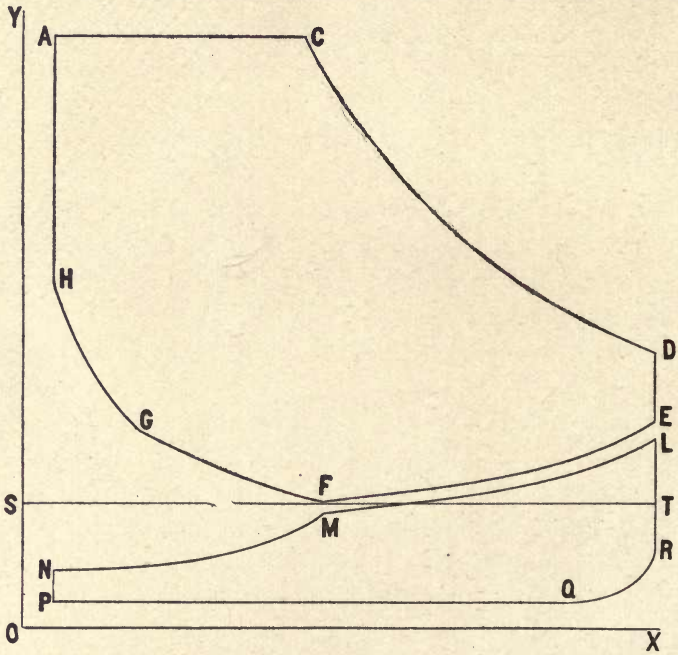


Fig. 26.

cylinder. The back pressure line E F, of the high pressure cylinder is parallel to the steam line L M, of the low pressure cylinder. They would coincide if there were no resistance to the flow of steam and no heat loss. The flow of steam from the high to the low pressure cylinder is cut off at F and the steam remaining in the high pressure cylinder is compressed in the cylinder and in the pipes. The exhaust closes at G and steam is compressed in the small cylinder from G to H.

Cut off occurs in the low pressure cylinder at M, and the steam in the cylinder expands, the curve M N being an equilateral hyperbola. Release takes place at N and the pressure falls to that of the condenser. The back pressure line P Q and the compression curve Q R are like those of a single engine.

The diagrams shown in Fig. 27 are from a Woolf engine. The lines are similar to those of the theoretical diagram shown in Fig. 26.

With compound engines of the Woolf type the steam passes

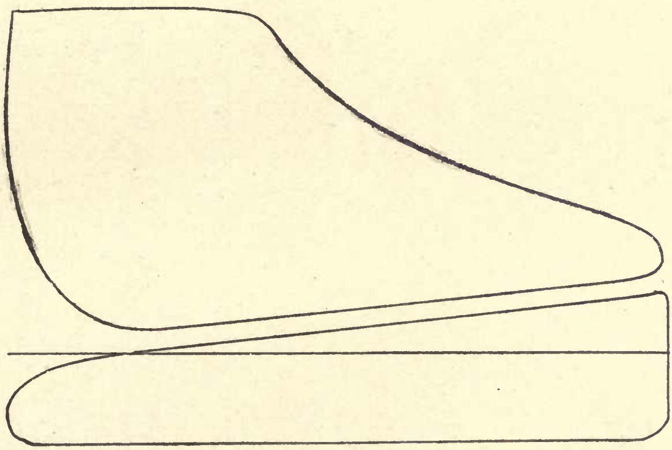


Fig. 27.

directly from the high pressure cylinder to the low pressure. In the **cross** compound, the cranks being at 90°, the piston in one cylinder is at the *end* of the stroke when the other is at the *middle* of the stroke. Therefore, a **receiver** must be used. The ideal cards from a cross compound engine are shown in Fig. 28. The dimensions are as follows:

Volume of high pressure cylinder	=	5 cu. ft.
Volume of receiver	=	5 cu. ft.
Volume of low pressure cylinder	=	15 cu. ft.
Cut off of high pressure cylinder	=	$\frac{1}{4}$ stroke.
Cut off of low pressure cylinder	=	$\frac{1}{2}$ stroke.
Initial pressure (absolute)	=	120 lbs.
Back pressure (absolute)	=	3 lbs.

Clearance and compression are not considered. The stroke of the high pressure cylinder begins at A. Steam is cut off at $\frac{1}{4}$ stroke at C.

Since steam is cut off at $\frac{1}{4}$ stroke, the ratio of expansion in the high pressure cylinder is 4. Then the volume of steam at D is 4 times that at C and since $p\ v = p_1\ v_1$, the pressure at D is $\frac{120 \times 1}{4} = 30$ pounds. Steam is released at D and passes to the receiver. The pressure at release, L, of the low pressure cylinder is found from the equation $p\ v = p_1\ v_1$. The volume of steam in the high pressure cylinder at cut off, is $\frac{5}{4}$ cubic feet and the

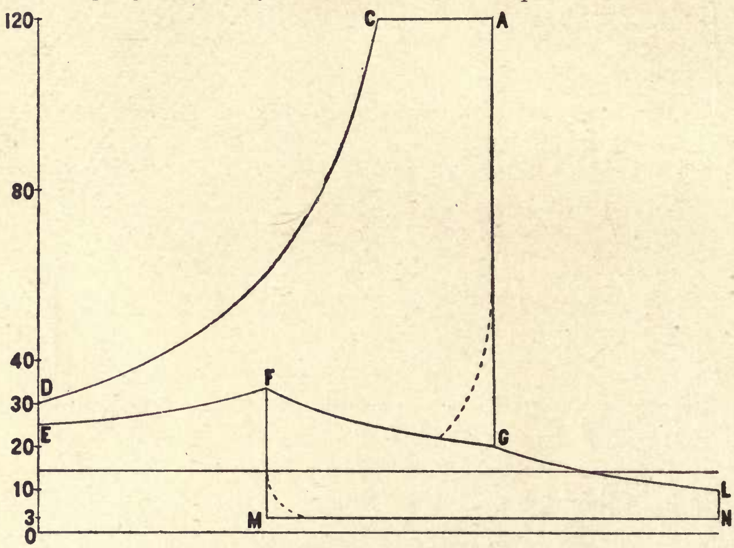


Fig. 28.

volume at L is 15 cubic feet; then the total ratio of expansion is $15 \div \frac{5}{4} = 12$. Then the pressure at L is $\frac{120 \times 1}{12} = 10$ pounds.

Since the low pressure cylinder cuts off at $\frac{1}{2}$ the stroke the volume at G is $\frac{1}{2}$ that at L.

Then as $p\ v = p_1\ v_1$, $p \times \frac{1}{2} = 10 \times 1$, or $p = 20$ pounds

Hence the pressure at G is 20 pounds.

We can now find the pressure at E. The pressure in the receiver when the low pressure cylinder cut off, was 20 pounds because the low pressure cylinder was in communication with it.

Then as the piston in the small cylinder is at the end of the stroke when the steam is cut off in the large cylinder, steam at 30 pounds pressure rushes from the high pressure cylinder into the receiver and mixes with steam at 20 pounds pressure. When steam is cut off in the low pressure cylinder the high pressure cylinder and the receiver are in communication with each other; the total volume being the volume of the high pressure cylinder plus that of the receiver or $5 + 5 = 10$ cubic feet. What we wish to find is the pressure of the steam of this volume. The formula is,

$$P\,V = p\,v + p_1\,v_1.$$

Since there are 10 cubic feet in the receiver and high pressure cylinder, the value of V is 10. The volume of steam in the high pressure cylinder is 5 cubic feet and its pressure is 30 pounds. The volume of steam in the receiver is 5 cubic feet and its pressure is 20 pounds, then,

$$10\,P = 5 \times 30 + 5 \times 20$$

$$P = \frac{250}{10} = 25 \text{ pounds.}$$

Then the pressure at E $= 25$ pounds.

While the piston of the high pressure cylinder is on the return stroke, steam in that cylinder is compressed from E to F; the volume being 10 cubic feet. When the piston has completed $\frac{1}{2}$ the return stroke, the volume at F is equal to $\frac{1}{2}$ the volume of the high pressure plus the volume of the receiver or $\frac{5}{2} + 5 = 7\frac{1}{2}$ cubic feet.

Then with the formula $p\,v = p_1\,v_1$ we obtain,

$$p = \frac{25 \times 10}{7\frac{1}{2}} = 33.33$$

or the pressure at F is 33.33 pounds.

Then as the cranks are 90° apart, and the high pressure piston is at the middle of the stroke, the low pressure piston must be at the beginning of its stroke; or the pressure in the low cylinder is the same as that in the high and in the receiver, or 33.33 pounds. Since the stroke is beginning in the low pressure cylinder, steam is being admitted to it from the receiver and consequently the pressure in the large cylinder falls as the volume increases, which

is shown by the line F G. Cut-off occurs in the low pressure cylinder at G, and the steam expands until the piston reaches L; the curve being an equilateral hyperbola. At L the release occurs and the pressure drops to that in the condenser; in this case about 3 pounds. The back pressure line is of course the pressure in the condenser.

The cards of Fig. 29 are from a cross compound engine. The rise of pressure in the middle of the back pressure line of the

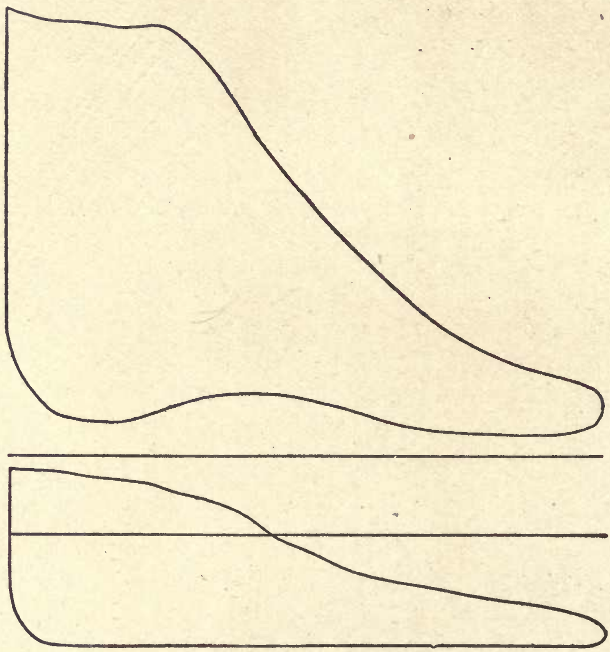


Fig. 29

diagram from the small cylinder is due to the fluctuation of pressure in the receiver.

COMBINED DIAGRAMS.

The indicator cards of multi-cylinder engines may be combined so that the pressures and volumes are shown in their proper relations. To do this, the cards are reduced to the same scale of pressure and the same scale of volume. To make the combined diagram of convenient size, the length of the low-pressure card is

left as it is and the length of the high-pressure diagram is shortened in the ratio of the cylinder volumes.

Perhaps the best way to show the method is by an illustration. The cards shown in Fig. 30 were taken from a compound condensing engine.

Ratio of cylinder volumes	= 1 : 3
Initial pressure	= 138 pounds
Spring, high-pressure card	= 60 pounds
Spring, low-pressure card	= 30 pounds

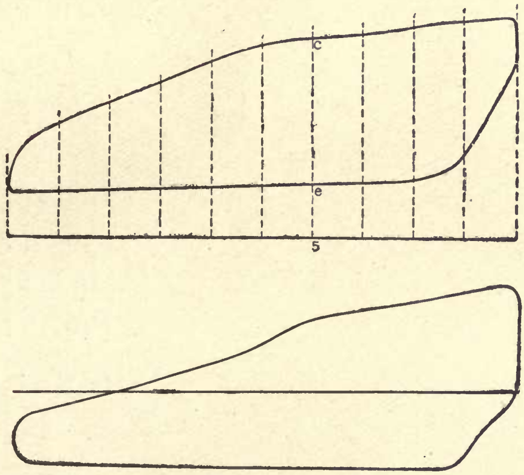


Fig. 30.

First draw the line of zero pressure A B and the line of zero volume E F (see Fig. 31). The line O P is drawn parallel to E F and at a distance proportional to the clearance of the low-pressure cylinder. Similarly the distance between M N and E F represents the clearance of the high-pressure cylinder. Now draw the atmospheric line C D. In this case it will be .49 inch above A B because a 30-pound spring is used, and $\frac{14.7}{30} = .49$. Reproduce the low-pressure card without change, as shown.

Divide the high-pressure card with 10 (or more) ordinates, and reproduce it with volumes and pressures of the same scale as the low-pressure card. Since a 60-pound spring was used, each ordinate will be twice as long (because $\frac{60}{30} = 2$). The distance

between the ordinates will be $\frac{1}{3}$ as great because the high-pressure cylinder is $\frac{1}{3}$ the volume of the low. Draw the ordinates as shown. The distance between ordinates L and M will be $\frac{1}{3}$ of the length of the high-pressure card. The points *c* and *e* are on the fifth ordinate, and are twice as far above the atmospheric line as they are in the original high-pressure card.

After locating all the points draw the curve through them. Now draw the theoretical expansion curve R S through the point of cut-off of the high-pressure cylinder by the method explained

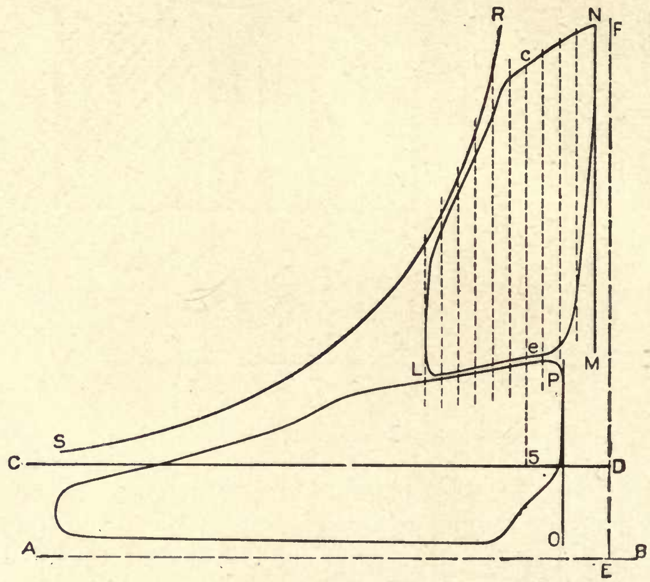


Fig. 31.

on page 39. The difference between the area included between the theoretical curve and the lines of no pressure and no volume, and the sum of the actual areas, represents approximately the losses.

This method is not strictly accurate, because all the steam used in the high-pressure cylinder does not pass to the low-pressure cylinder: a small portion is left for compression. By thus combining the cards the action of the valves may be discussed, provided such data as size, type and speed are known.

The combined diagram of a multi-expansion engine drawn

according to the above method is shown in Fig. 32. The volumes and pressures are first reduced to the scale of the low-pressure card. The cards are then redrawn at the proper distances from the lines of no pressure and no volume; the clearance in each cylinder being considered.

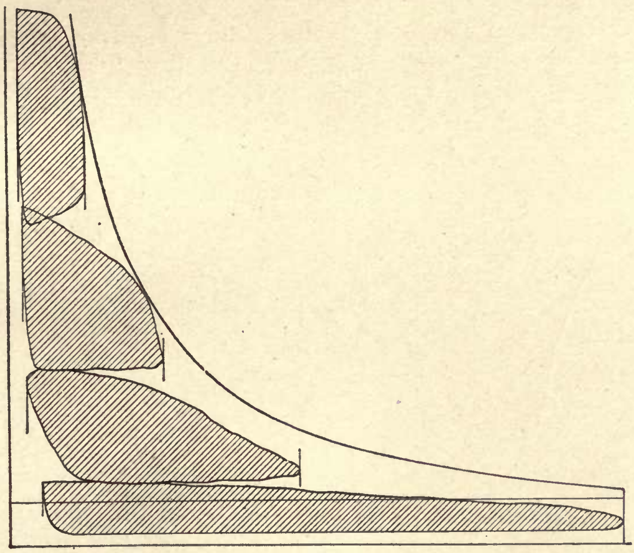


Fig. 32

HORSE-POWER OF COMPOUND ENGINES.

The I. H. P. of multi-cylinder engines may be found by adding the I. H. P. of the several cylinders. Another method is to reduce all the pressures to the area of the low-pressure cylinder. This is done by dividing the M. E. P. of each cylinder by the inverse ratio of its volume to that of the low-pressure cylinder.

Suppose the M. E. P. of the high-pressure cylinder of a compound engine is 78 pounds as found from the indicator card. If the volume of the high-pressure cylinder is $\frac{1}{3}$ that of the low, the M. E. P. of the high referred to the low would be $\frac{78}{3} = 26$ pounds. Then if the card from the large cylinder shows a M. E. P. of 30 pounds, the total M. E. P. is $30 + 26 = 56$ pounds, and the I. H. P. is found by inserting 56 as the value of P and using the area of the low pressure as A, in the formula for I. H. P.

INDICATOR CARDS.

The indicator reveals defects in the steam distribution. That is, if the valve or valves are set so that the events of the stroke are *too early* or *too late* or if *more work* is done at one end of the cylinder than at the other, the engineer finds it out by examining the indicator card.

Sometimes an engine appears to run well and the owner is perfectly satisfied with it; but from the indicator diagrams it is found that considerable saving might be effected by correcting the defects in the valve setting.

On looking at a diagram one might say it was a faulty card

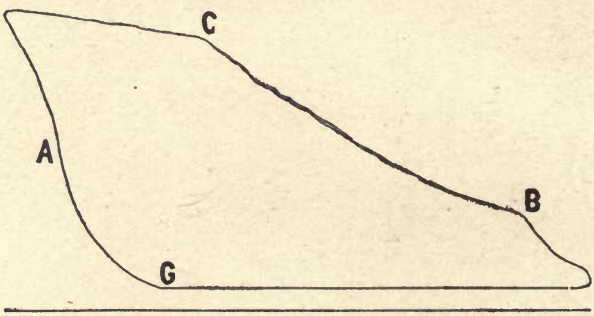


Fig. 33.

and yet for that type, size and speed it is perhaps the best that could be obtained from the engine. The same form of diagram is not possible from different types of engines. The diagram from a Corliss engine, having four valves, is different from that of the plain slide valve engine; also the diagrams from high speed engines differ from those of low speed.

The most common faults in the distribution of steam in the cylinder can be divided into four classes.

a = **Admission** too early or too late.
b = **Cut-off** too early or too late.
c = **Release** too early or too late.
d = **Compression** too early or too late.

In the following figures,

A is the point of admission,
C is the point of cut off,
B is the point of release,
G is the point of compression.

The diagram shown in Fig. 33 shows **too early admission.** The admission line curves *backward* instead of being straight and perpendicular to the atmospheric line as it is in Fig. 24. The diagram also shows cut off, release and compression early. When the valve is of the plain slide valve type *all* the events are likely to be too early if one of them is. The reason for the event being

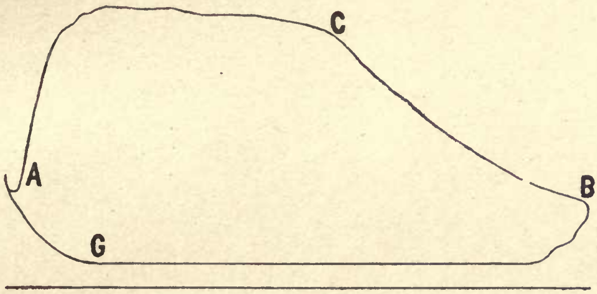


Fig. 34.

too early is that the eccentric has too much *angular advance.* Hence the remedy is to decrease the angular advance.

Fig. 34 shows a diagram having the opposite defects to those of Fig. 33. The events are too late. The admission line curves forward and the line shows that admission does not take place until after the stroke is well begun. Release occurs at the end of

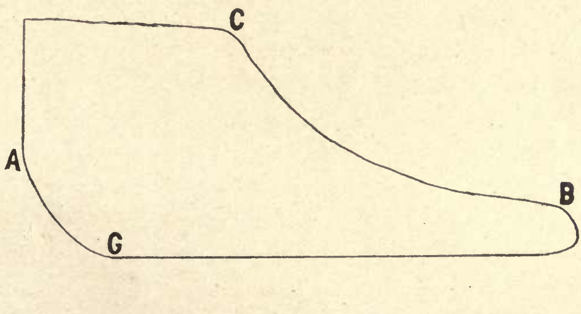


Fig. 35.

the stroke. In this case the angular advance of the eccentric should be increased until the admission line is perpendicular to the atmospheric line.

Fig. 35 shows a card having too much back pressure. This may be due to a small exhaust port or pipes, or the passage of steam

through coils of pipe for heating. The card shows the other events to be good. A diagram having **too late cut off** is shown in Fig. 36. The pressure at release is high. When the engine is running under this condition much of the benefit from expansion is lost.

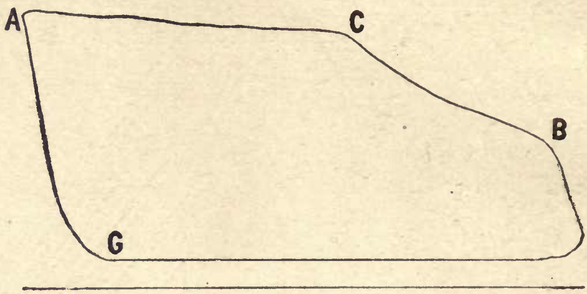


Fig. 36.

A diagram from a condensing engine is shown in Fig. 37. These oscillations are caused by the vibration of the indicator piston and spring. To avoid these vibrations never use a very light spring for high speed.

The diagram of Fig. 21 shows **too early cut off.** In this case the cut off is so early that the expansion line extends *below* the

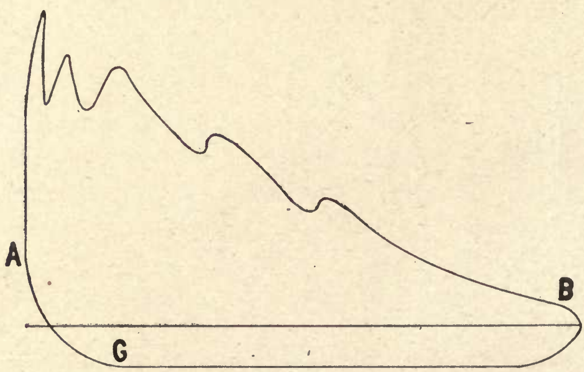


Fig. 37.

atmospheric line making a loop. The area of the loop must be *subtracted* from the area of the card as explained on page 33.

Fig. 38 shows a pair of diagrams from a plain slide valve engine. The admission lines are good. The sloping steam lines

show wire-drawing due to the slow action of the valve or too small ports or pipes. This wire-drawing decreases the area of the card which indicates loss. The greatest fault is the inequality of area of the diagram for the ends. The late cut off and consequent late compression of one end causes more area than the too early cut off and too early compression of the other end. These

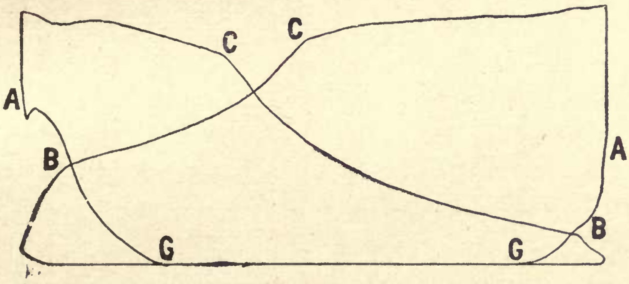


Fig. 38.

cards can be improved by adjusting the angular advance of the eccentric and the length of the valve rod.

The diagram of Fig. 39 indicates too early compression. The compression curve extends above the initial pressure line. The area of the loop must be subtracted from the card area, when com-

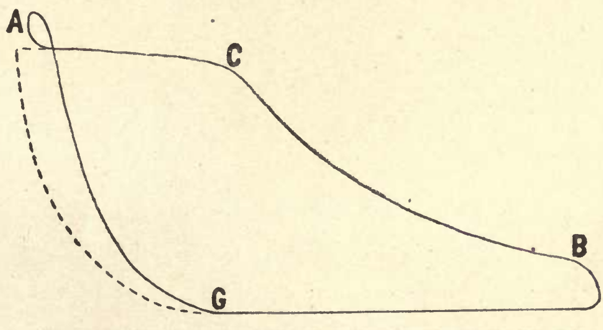


Fig. 39.

puting the I. H. P. If the cut off is kept the same and the compression made what it should be, the gain in area would be the area included between the full line and the dotted line plus the area of the loop. The remedy for this case is to decrease the inside lap.

The amount of compression varies with the speed and type of engine. Slow speed engines require *less* compression or cushioning than high speed. The exhaust steam should *never* be compressed higher than the boiler pressure.

For high speed engines the compression should extend to about .9 the initial pressure. For medium speeds about .5 and for low speed .2 to .4.

In the case of a slide valve engine it is not always possible to set the valve so that the card may have all the events as they should be. Sometimes the laps of the valves should be altered. For too much compression decreases the inside lap. For too early cut off decrease the outside lap.

If the valve travel is increased, compression is retarded, that is, decreased; release occurs sooner.

STEAM CONSUMPTION.

The amount of steam used by an engine is called its **Steam consumption** and for comparison, it is customary to state the amount of steam consumed per indicated horse-power per hour. By means of the indicator diagram the steam consumption can be computed approximately.

To find the Steam Consumption from the Diagram. The diagram shown in Fig. 40 is from a 20 × 36 engine, running at a speed of 80 revolutions per minute. A 40-pound spring was used.

By measuring the card, we find the mean ordinate to be .91 inch and the M. E. P. $= .91 \times 40 = 36.4$ pounds.

$$\text{I. H. P.} = \text{Engine Constant} \times \text{M. E. P.} \times \text{piston speed.}$$

$$= .00952 \times 36.4 \times 480.$$

$$= 166.33.$$

In Fig. 40 L M is the atmospheric line and O R the line of zero pressure drawn so that O L = 14.7 pounds. O N is the clearance volume = 8 per cent of the piston displacement. The line P Q is drawn from O R to some point on the compression

curve. From D, a point on the expansion curve before release, the line D F is drawn perpendicular to O R.

Then from the diagram,

O R = 3.24 inches.
O F = 3.00 inches.
O P = .345 inch.
D F = .795 inch.
P Q = .795 inch.

The length of stroke is 36 inches or 3 feet, and the length of the diagram 3 inches. Then 1 inch of the length of the card corresponds to 1 foot of the stroke. The scale of spring used is 40. Therefore we can easily reduce the above dimensions to pounds pressure and to feet.

O R = 3.24 feet.
O F = 3.00 feet.
O P = .345 feet.
D F = 31.80 pounds.
P Q = 31.80 pounds.

The area of the piston (head end) is,

$$\frac{\pi d^2}{4} = \frac{3.1416 \times 400}{4} = 314.16 \text{ sq. in.} = 2.18166 \text{ sq. ft.}$$

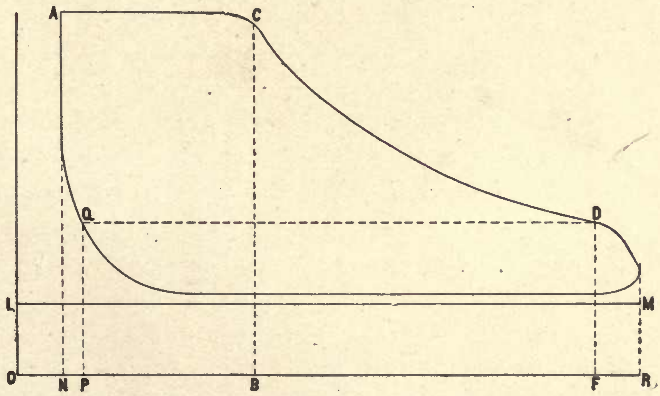


Fig. 40.

We can now find the volume of steam at any point of the stroke.

When the piston is at D, the volume is,

$$2.18166 \times 3 = 6.54498 \text{ cubic feet.}$$

When the piston is at Q, the volume is,

$$2.18166 \times .345 = .75267 \text{ cubic foot.}$$

From the steam tables we can find the weight of a cubic foot of steam at a given pressure.

The weight of 1 cubic foot at 31.8 pounds absolute pressure is .07773 pound. Then the weight of steam present when the piston is at D is,

$$6.54498 \times .07773 = .50887 \text{ pound.}$$

The weight of steam present when the piston is at Q is

$$.75267 \times .07773 = .05852 \text{ pound.}$$

The weight of steam in the cylinder is .50887 pound and the weight of steam kept for compression is .05852 pound. The weight exhausted per stroke is,

$$.50887 - .05852 = .45035 \text{ pound.}$$

The amount used per I. H. P. per hour is,

$$\frac{.45035 \times 2 \times 80 \times 60}{166.33} = 25.993 \text{ pounds.}$$

This may be stated in words as follows:

Measure the pressure from the vacuum line to some point in the expansion curve before release and from the steam tables find the weight of a cubic foot at that pressure. Multiply the volume (in cubic feet) of the cylinder, including clearance to that point, by the weight per cubic foot. The result is the weight of steam in the cylinder. As this weight includes the steam used for compression it must be corrected to obtain the weight used per stroke.

Take some point on the compression curve and measure its absolute pressure. Then compute the weight of steam to this point. Subtract this weight from the weight of steam to the point in the expansion curve and the result is the weight of steam used per stroke.

Multiply the weight of steam used per stroke by the number of strokes and divide by the indicated horse-power as found from the card. The final result is the number of pounds of steam consumed per horse-power per hour.

We may also calculate the steam consumption by taking the point of the expansion curve near the cut-off.

$$O\,B = 1.23 \text{ inches} = 1.23 \text{ feet of the stroke.}$$

$$B\,C = 1.8 \text{ inches} = 72 \text{ pounds.}$$

The weight of 1 cubic foot of steam at 72 pounds absolute pressure is .1671 pounds. Then the volume of steam at C is,

$$2.18166 \times 1.23 = 2.68344 \text{ cubic feet.}$$

The weight at C is,

$$2.68344 \times .1671 = .44840 \text{ pound.}$$

The weight kept for compression is the same as previously found, *i. e.*, .05852 pound.

Then the weight of steam used per stroke is,

$$.44840 - .05852 = .38988 \text{ pound.}$$

The steam consumption per I. H. P. per hour is,

$$\frac{.38988 \times 2 \times 80 \times 60}{166.33} = 22.50 \text{ pounds.}$$

If the valve doesn't leak, the amount of steam just after cut off should equal the amount just before release, but our calculation shows that there is .45035 — .38988 = .06047 pound more at release than at cut off. This shows that at entrance .06047 pound was condensed before the piston reached C and was re-evaporated before release. This calculation gives an idea of the amount of cylinder condensation.

If there is considerable compression as in Fig. 40 the above method may be simplified by taking the two points D and Q at the same height above the vacuum line. The pressures will then be the same.

Let W = weight of steam used per I. H. P. per hour.
w = weight of one cubic foot of steam at the absolute pressure D F.
L = length of the diagram, N R.
l = distance between Q and D.
P = mean effective pressure.

Then,

$$W = \frac{13{,}750 \times w \times l}{P\ L}.$$

Example. A card was taken from a 12 × 14 engine. Length of card L = 3.5 inches, l = 2.875 inches. Absolute pressure at

D = 30 pounds. M. E. P. = 30.75 pounds. What is the steam consumption per I. H. P. per hour?

$$W = \frac{13{,}750\, w \times l}{P \times L}$$

$$= \frac{13.750 \times .0736 \times 2.875}{30.75 \times 3.5}$$

$$= 27.03 \text{ pounds.}$$

This method gives only an approximate steam consumption On account of the leakage of valves and the initial condensation of steam in the cylinder, the actual consumption is somewhat greater. The excess varies considerably and makes all results so obtained of little value. In our calculation we used one diagram only, that for the head end; we assumed the diagram from the crank end to be the same. The indicated horse-power for each end should be computed, as it is subject to considerable variation. In the above formula the average mean effective pressure should be used. The results of calculations of steam consumption from indicator cards are so unreliable that engineers place little dependence upon them. The results may be used as a basis for estimates, but for accurate knowledge, an engine test must be resorted to

EXAMPLES FOR PRACTICE.

1. Given the following data to find the M. E. P. Area of card 2.79 square inches, length of card 3.1 inches, scale of spring 50. Ans. 45 pounds.

2. Steam enters a cylinder at a pressure of 210 pounds by gage and leaves at 3 pounds pressure (absolute). What is the thermal efficiency? Ans. 29 per cent.

3. An engine has a B. H. P. of 78.96. The I. H. P. is 86.73. What is the mechanical efficiency. Ans. 91 per cent.

4. An engine develops 149.97 I. H. P. If the piston diameter is $17\frac{3}{4}$ inches, the stroke 30 inches, and the M. E. P. 40 pounds, what is the speed? Ans. 100 revolutions per minute.

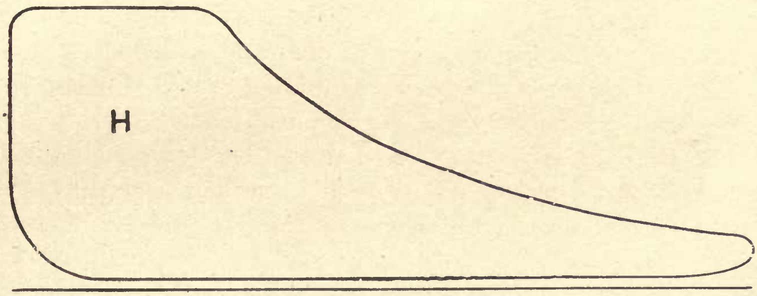
5. The theoretical M. E. P. of a triple expansion marine engine is 48 pounds. What is the probable actual M. E. P.?

6. The cylinders of a triple expansion are 12, 30, and 75 inches in diameter respectively. The stroke is 24 inches. The

mean effective pressures from the cards were 118.4, 51.6 and 19.34 pounds respectively. What is the I. H. P. when the speed is 125 revolutions per minute? Ans. 2,050 I. H. P.

7. Find the steam consumption from the following card. The engine was running at 75 revolutions per minute, and developing 230 I. H. P. when the card was taken. A 60-pound spring was used, and the M. E. P. for this end was 46 pounds. Assume crank-end card to be the same as the head end.

Ans. 22 pounds (about).



Suggestion: First draw a line .3 inch above the atmospheric line, then reduce this ordinate to *absolute* pressure, and use formula at the bottom of page 55.

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